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--Office of Production
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Solar energy utilization
for house heating.

RESEARCH REPORT

SOLAR ENERGY UTILIZATION

FOR

HOUSE HEATING

OFFICE OF
RODUCTION RESEARCH AND DEVELOPMENT
WAR PRODUCTION BOARD
WASHINGTON, D. C.

Laboratory Work Conducted By Engineering Experiment Station University of Colorado

Work Coordinated By Industrial and Consumer Products Branch

Date May 18, 1946

Report No. Project No. 459 Contract No. 100

Copy No.

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May 18, 1946

Mr. Ely C. Hutchinson
Production Research and Development Division
Office of Declassification and Technical Services
Department of Commerce
Washington 25, D. C.

Dear Mr. Hutchinson:

In accordance with the requirements of contract WPB-100, research and development work has been conducted on the Solar Heat Trap proposed by the Consumer Products Branch, Office of Production Research and Development of the War Production Board. The details of this investigation and the results thereof are hereby submitted ~~in~~ the accompanying Final Report.

Very truly yours,

George O. G. Löf

George O. G. Löf
Associate Professor of
Chemical Engineering

GOGL:LJ

SOLAR ENERGY UTILIZATION

for

HOUSE HEATING
(Contract WPB-100)

University of Colorado
Engineering Experiment Station
Boulder, Colorado
May 18, 1946

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ACKNOWLEDGEMENT

In most research projects having a rather broad scope in a relatively unexplored field, numerous workers may each contribute a part of the complete solution. Such has been the case in the performance of the research covered by this report. The authors, therefore, wish to acknowledge the excellent work of those without whose services this research could not have been conducted.

The performance of a large part of the preliminary testing and the initial studies on the indoor unit were very ably handled by Mr. Morton David. His successor, Mr. Jack Pohlenz, assisted in the carrying of this work to its conclusion, and the developing of data for construction of the outdoor laboratory unit. The junior authors of this report then aided in the construction and operation of the full scale laboratory unit, aided in designing the house unit, and performed the calculations of their performance.

Particular recognition is given to the services of Professor C. H. Prien of the Chemical Engineering Department, who was in active charge of the experimer al work on the indoor unit and the outdoor laboratory unit during most of the period of their operation. He was responsible for securing the major portion of the data on which the results are based, and numerous features of construction and techniques of operation were developed under his direction.

Thanks are also accorded to Professor W. B. Pietenpol, Professor of Physics, and Professor B. H. Spurlock, Assistant Professor of Mechanical Engineering, who, with the senior author, constituted the committee which planned initial phases of the work. Advice and encouragement received from Professor C. W. Borgmann, Direc or of the Engineering Experiment Station, have been greatly appreciated.

The authors also wish to express their appreciation for the assistance given by members of the Geology Department of the University of Colorado, and by the technical personnel of the Boulder and Valmont offices of the Public Service Company of Colorado. The information on various phases of the weather and house heating requirements proved invaluable. Thanks are also tendered to Mr. I. F. Har and his associates of the Solar Radiation Section of the U. S. Weather Bureau for their calibration of the pyrheliometer and for considerable data concerning its use.

To Mr. K. W. Miller, former consultant to the OPRD, who devised this principl of solar energy collection, and to Mr. Ely C. Hutchinson, chief of the Consumer Products Branch of the OPRD during most of the period of this study, go the prais of the authors for their excellent services to the people of the United States, and their continued interest and advice concerning the performance of this work.

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I SUMMARY

The work described in this report was undertaken at the request of the Consumer Products Branch of the Office of Production Research and Development of the War Production Board. The specific problem was the experimental investigation of a proposal advanced by K. W. Miller (10), consultant to the OPRD, for the practical utilization of solar energy, principally for purposes of house heating.

Miller's proposal involved a design consisting of a group of partially blackened, overlapping glass plates, mounted on a house roof. The arrangement is similar to that of shingles, separated by small air spaces. A semi-continuous glass cover over the staggered plates was also proposed. By virtue of the high transmissivity of glass for solar radiation and the low transmissivity for long wave thermal radiation, the black and clear glass surfaces in the unit would become heated when exposed to sunlight, and the reradiated heat therefrom would have no avenue of escape. Air passing between the plates could then be heated and supplied to the house. By this method, it was claimed that heat could be collecte at a higher temperature level than is attained in other types of collectors.

A second proposal, not covered by contract nor studied experimentally, concerned the storage of the collected heat by passing the hot air through a bed of loosely packed, cheap solids, to which the sensible heat of the air could be transferred. Delivery of hot air from the storage bed could then provide heat when no solar energy was being received.

The first experimental unit was constructed indoors and irradiated artificiall with a bank of tungsten lamps. In a complete study of the operating characteristics of this unit, it was found that Miller's principle was actually workable and that a construction involving two-thirds overlap of plates blackened one-third their length on the upper surface and spaced 1/4 inch apart resulted in the best performance. Reports covering this phase of the investigation have been previousl presented. (11, 12, 13, 14, 15)

The results obtained with the indoor unit were next used in the design of a large scale collector, which was constructed on the laboratory roof. This unit was operated intensively over a three-month period in the fall and run intermittent ly throughout the winter. The large number of variables measured and correlated included the solar heat input, heat collected, efficiency of collection, entrance and exit air temperatures, air flow rate, plate spacing, and various temperatures needed for establishing heat balances. The principal results of the investigation were obtained in this way.

A third unit was then constructed on a small house and connected by ducts to the regular hot air heating system. This collector covered about one-third of the entire roof area and occupied the usable surface on the south side of the roof Electrical connections and automatic controls completed the installation and permitted its completely automatic operation. By means of this installation, the numerous problems associated with the combination of the solar heating and normal heating system were solved and the fuel savings effected by the solar unit were measured throughout one winter. A system, designed to utilize the solar heated air to supply hot water for household use in the summer, was also installed and operated for a short period.

From the data experimentally obtained in tests on the three units, numerous results were calculated; several predictions, not based specifically on experimental data, were made. The latter included principally those concerning heat storage, operation of the equipment in other localities, air conditioning, and cost.

The results show that Miller's principle, although too idealized, is workable, and that by it solar heat can be recovered with efficiencies of 35 per cent to 40 per cent, with exit air temperatures above 150°F at noon on a bright day. Higher temperatures can be secured, but at lower efficiencies. Tilting of the collector to the south is desirable, and roof slopes ranging from the one equal to the latitude, up to approximately 10 degrees greater than the latitude, are near the optimum. Breakage of glass plates exposed to solar radiation, remains the greatest problem. The way in which thermal stresses cause this breakage is subject of a new investigation.

Operation of the house unit during one winter showed that a fuel saving of about 20 per cent was being realized, even though ideal arrangements had not yet been made. Very satisfactory automatic equipment controlled the unit, and the collector showed excellent weather resistance. Calculations show that if the heat load carryable is to be increased, the ducts will have to be insulated; and to secure a sizeable change, storage of heat for a maximum economical period of one day should be provided. Under these circumstances, and with the roof area employed, 50 per cent to 60 per cent of the fuel used in the test house could be saved. A storage bed containing about 6 tons of solids would suffice, and if air recirculation between collector and storage unit were provided, the storage bed might be appreciably smaller.

Very limited data indicate that the house hot water can be supplied by the solar water heater during the summer.

Locations north of the 40th parallel are generally unfavorable for economical solar house heating. Because of the possibility for using the solar heated air to operate an absorption type of air conditioner, it is felt that the greatest applicability of this system would be in the southern belt of the states. It is also believed that the net cost of the solar heating apparatus could be kept below $500 on large scale production.

The greatest advantages of this type of solar heat collector are its simplicity of construction and low cost. It appears to have an efficiency in the same range as that of the flat plate collectors in which water is heated (9), and its operation is more efficient and flexible than the method employing large south windows.

The outlook for commercial development of this apparatus is highly favorable. Work being sponsored by a private manufacturing company is now under way at the University of Colorado. It is expected that this work will lead to (1) the development of a practical design suitable for factory production of solar heating units, (2) the determination of the most satisfactory means of heat storage, and (3) the elimination of plate breakage.

Applications for patents on the basic principles and the improvements have been made by K. W. Miller and G. O. G. Löf, and licenses to the government have been granted. A copy of the Löf patent application has been included in this report to show the design of the entire apparatus in its present stage of development.

II INTRODUCTION

During the last 70 years many attempts have been made to bring about more effective utilization of the great quantity of solar-energy incident on the earth' surface. A few of the names connected with these attempts to prove the economic feasibility of converting solar energy to heat and transferring that heat to the working fluid of a heat engine, or storing it in some suitable fluid for subsequent utilization are: Mouchot (1, 2), Pifre (3, 2), Shuman (4), Ericsson (5), Willsie (6), Shuman (7), Abbot (8), and Hottel and Woertz (9).

Solar heat collectors might be classified in three ways; according to the type of insulation used, the degree of concentration of sunshine obtained, and the nature of the orientation of the collector with respect to the sun. Insulation of the heat collection surface is accomplished in several ways. In the flat plate collector the insulation may consist of one or more spaced glass panes placed parallel to the absorbing surface and the sides and bottom could be covered with a layer of a commercial insulating material. In the case of the tubular collector one or more concentric glass tubes may surround the heat absorbing tube and in some cases a vacuum is maintained between the tubes. In the flat plate collector there is usually no concentration of the suns rays but in other types of collector reflectors are sometimes employed. Any collector might be built with three types of orientation. It might be mounted permanently in one position so selected to gather the maximum amount of solar energy over the desired period; to improve its efficiency the unit might be mounted so that its angle of inclination might be varied day to day so that it would always be perpendicular to the suns rays; further elaboration may be used in the mounting to permit the unit to follow the hour angle of the sun so that it is at all times aligned in the position to obtai the maximum amount of radiant energy from the sun.

In the design of a solar heat collector, two factors of major importance must be considered. One is the quantity of heat available at the earth's surface, and the other is the temperature level at which it is collected. Mirrors which focus the heat of sunlight and allow it to be collected at relatively high temperature levels have been devised but these have a disadvantage in that they collect only a small portion of total energy. On the other hand large quantities of heat can be collected at low temperature levels by the "greenhouse effect". This effect utilizes the fact that glass is transparent to visible and red wave lengths which make up the predominate portion of sunlight and is opaque to long infrared wave lengths. When the sunlight enters the greenhouse, it impinges on the contents an is changed to low temperature long wave length heat which is trapped inside the glass enclosure.

Since focusing devices of large area are quite expensive to build and to operate it seems desirable to devise a method to utilize the greenhouse effect in such a way that the temperature level of collection is greater than previous designs have allowed.

A modification of the greenhouse effect was suggested by K. W. Miller (10) of the Office of Production Research and Development of the War Production Board by which he proposed to raise the temperature level of the heat collected to several hundred degrees Fahrenheit. This plan involves the use of a large area covered by overlapping plates of glass. The area of each sheet of glass, which extends in

the space on the under side, is coated with a radiation absorbing medium such as lamp-black. Air is drawn slowly through the space between the plates, becomes heated, and is removed from the unit.

It is the purpose of this investigation to ascertain the workability of the scheme presented by Mr. Miller and to determine the optimum dimensions and configuration of the heat collecting unit. After it was found that the scheme offered an improvement over previous methods of solar heat collection, it was decided to place a unit in operation as an auxiliary heating unit in a private residence to determine its economic feasibility.

The following report describes in detail the work done and the results obtained during the course of the investigation with the exception of the preliminary tests on an experimental indoor heat collector. The latter is covered in previous reports (11, 12, 13, 14, 15).

III THEORY

The theory of the solar heat trap as proposed by K. W. Miller will be described in connection with Figure I which has been modified slightly from Miller's original drawing.

A large area is covered by overlapping sheets of glass and enclosed within heat insulating walls. Solar heat impinges on the upper glass, part of it is multiply reflected back towards the sky, part is absorbed in the glass, and the remaining transmitted portion impinges on the black surface of the opaque glass where it is absorbed and changed to heat. As the black surface absorbs heat its temperature rises and it reradiates in the far infrared region to which radiation glass is opaque.

Air is slowly drawn inward through or between the glass plates. This air at base temperature picks up the solar heat energy absorbed in the glass. This air is passing counterflow to the heat flow by conduction through and along the glass plates and by radiative heat transfer upward between them. Thus most of the heat attempting to escape upward from the high temperature end of the opaque glass plate is picked up by the air stream which, therefore, enters tho opaque glass section at an elevated temperature. Very important is the fact that this air flow maintains the upper surface of the glass plates at a temperature not much greater than air temperature. This reradiation of heat energy back toward the sky from the glass is thus very materially reduced since, quantitatively, it is proportional to the fourth power of the absolute temperature of the outer glass surface.

Mr. Miller has made a mathamatical analysis of the operation of the above described solar heat trap and has derived equations of use in the calculation of the possible heat recoveries and temperature levels of heat collection. The principal derivations made and the resulting final equations are shown below.

To analyse the existing conditions in a solar heat trap, steady state, temperature and air flow conditions must first be assumed. The calculations are made for a zenith position of the sun.

A heat balance is first established. A value for the absorption and reflection of a single sheet of glass is determined and from the following equations the reflection and absorption of the series of overlapping plates are found.

$$r_{n+1} = r_n + t_n^2 r/(1 - rr_n) \qquad (1)$$

$$a_{n+1} = a_n + t_n\,(ra_n + a)/(1 - rr_n) \qquad (2)$$

$$t_{n+1} = t_n t/(1 - rr_n) \qquad (3)$$

$$r_{n+1} + a_{n+1} + t_{n+1} = i = 1 \qquad (4)$$

where i = incident light or radiation
r = fraction reflected
a = fraction absorbed
t = fraction transmitted
n = subscript referring to number of plates

SOLAR HOUSE HEATING

DIAGRAMATIC SKETCH OF UNIT

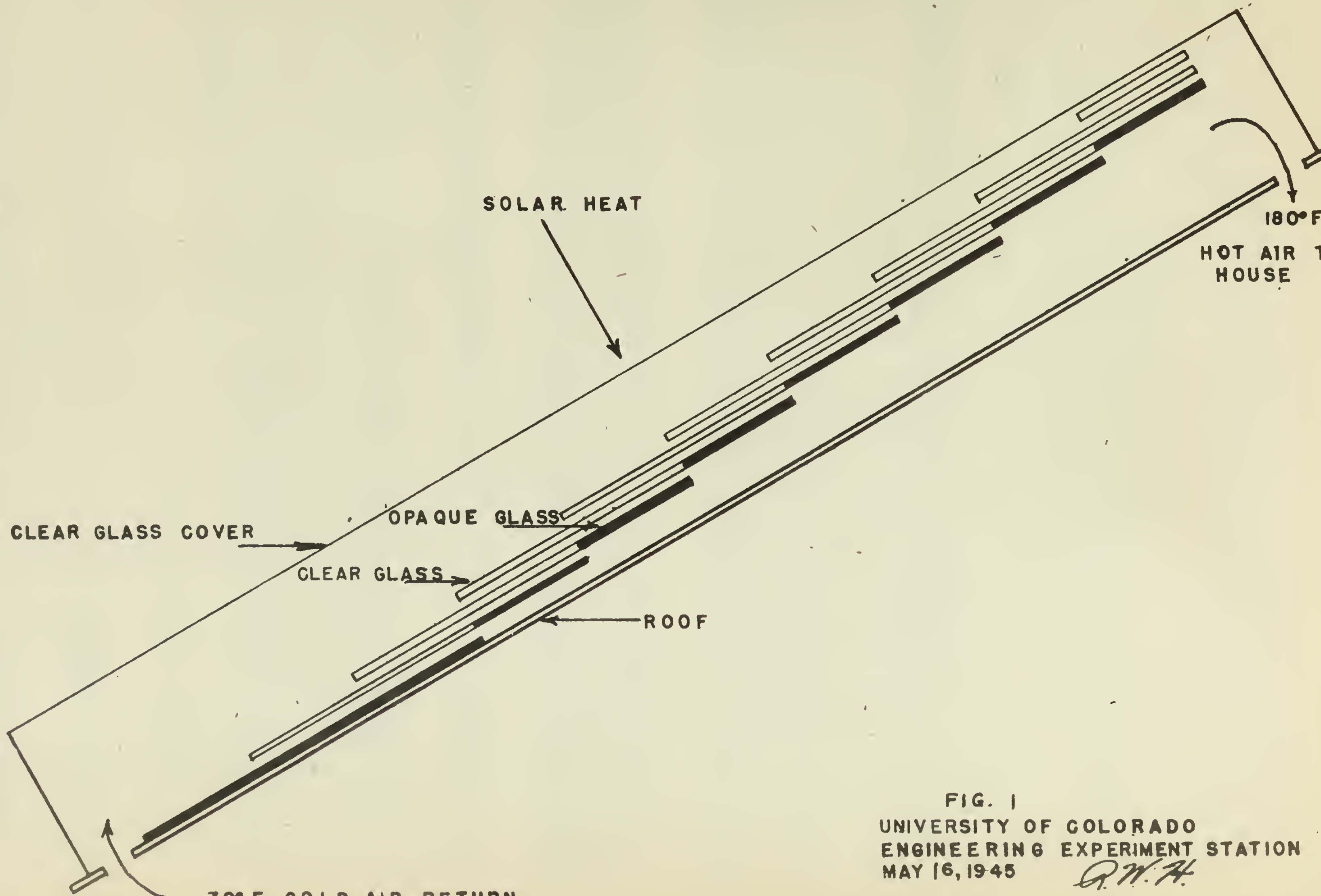


FIG. 1
UNIVERSITY OF COLORADO
ENGINEERING EXPERIMENT STATION
MAY 16, 1945 G.W.H.

Reradiation back to the sky from the top plate is found by assuming a temperature for the plate and applying the following formula:

$$Q = .172\ e\left(\frac{t + 460}{100}\right)^4 \qquad (5)$$

where e = emissivity of the glass plate (.95)

t = temperature of the top plate, °F.

In the above equation, it has been assumed that the surface is radiating to a sky having a temperature of absolute zero. It was found that a temperature of the surrounding air was a much better approximation.

An initial air temperature and a spacing between plates are first chosen in carrying out the following calculations.

A value is assumed for the ratio of actual spacing to critical spacing $s = \frac{2h}{2h_o}$, and a value is assumed for the heat flowing into a single air film (q)

By step wise procedure using the above relations, heat balances are set up around each plate and the distribution of absorbed radiation into the various plates and into the air stream are determined. Temperatures of air films are obtained from the simplified approximate equation:

$$T = \frac{-q}{16h^3k}(5h^2-y^2)(h^2-y^2) + \left[\frac{T_o(h+y) + T_{oo}(h-y)}{2h} + \frac{\Delta T x}{w}\right]$$

where

T = temperature at a point in the film
h = one-half the distance from one plate to the other
k = thermal conductivity
y = distance from center of air film to point
T_o = upper glass plate temperature
T_{oo} = lower glass plate temperature
ΔT = Temperature rise of glass plates from one end and the other of section taken
w = length of section taken
x = distance from center of section taken

The last equation and others to follow were derived by a series of approximations from the exact formula given below.

$$-\frac{dp}{dx} = \frac{1}{g}\frac{\partial}{\partial y}\Big[\ \text{[illegible]}\ \Big]$$

In the derivation of this equation, it was assumed that streamline motion of air between the plates prevails.

where p = pressure, #/ft²
g = acceleration of gravity = 32.17 ft/sec²
μ = viscosity of gas #/ft sec
C = heat capacity BTU/#°F

Average film temperatures are calculated from the formula:

$$\bar{T} = \frac{-17}{70} s(T_{oo} - T_o) + \frac{1}{2}(T_{oo} - T_o)$$

Now a new value of heat of conductivity is calculated at the mean temperature $\bar{T}$. Then $2h_o$ is calculated for each film by the equation

$$2h_o = \frac{2k(T_{oo} - T_o)}{q}$$

and new values of s are derived for each film by the equation

$$s = \frac{2h}{2h_o}.$$

Now by plotting $\bar{T}$ against x and extrapolating to $x = \pm \frac{w}{2}$ and plotting similar curves for the glass surface temperatures, T/w can be taken from the curves. Using this value and the value of $\Delta\bar{T}$ for each film, new values of q may be found by the relation.

$$q_a : q_b : q_c : \text{-----} = \Delta\bar{T}_a : \Delta\bar{T}_b : \Delta\bar{T}_c : \text{-----}$$

With the new values of q and s, new values of q_1 and q_2, the heat entering the air stream from each plate may be calculated and a new corrected heat balance set up.

With the above heat balance, the mass air flow can be calculated and from that the velocity of the air stream.

By the above analysis it was found that exit air temperatures of 300°F. could easily be obtained in a unit built as described and shown in Figure 1. It was assumed, however, that no convection losses took place at the cover plate surface and that no heat was lost by conduction through the sides or floor of the unit. These assumptions, along with the basic one that streamline flow prevails, make the actual predicted numerical results questionable even though their general order of magnitude may be correct.

IV CONSTRUCTION AND OPERATION OF SOLAR UNITS

I. General Procedure

The investigation of the collection and utilization of solar heat by the method which involves the use of overlapping glass plates as suggested by K. W. Miller was taken up in three phases. The first phase consisted of building an experimental collection unit and an artificial light source indoors. Tests on this unit were to furnish such information as general workability, optimum plate spacing, optimum overlap, optimum plate length, and optimum air rate.

The data made available through the indoor unit studies were used in the design of a larger heat collection unit which was constructed on the south roof of the laboratory, and the data were checked under natural sunlight. The effect of the changing solar angles was studied along with the effect of clouds and haze. Exit air temperature and overall heat recovery efficiency were correlated with air rate and other operating variables.

These two phases being completed, the final phase of the investigation was to place a typical unit in operation in a local home and determine its usefulness in supplying heating requirements.

II. The Indoor Experimental Unit--Summary

A. Construction

The indoor solar heat collection unit was built for the purpose of obtaining design information to aid in the selection of the proper glass plate arrangement in the proposed outdoor unit. A drawing of the completed unit is shown in Figure I of "Progress Report V" of the Solar Radiation Investigation (15) and a description is also given. Only a brief summary is given here because the work with the indoor unit yielded only preliminary design results

The unit consists of a rectangular box 2-1/2 feet wide, 12 feet long, and 1-1/2 foot deep, well insulated against heat loss. Air at room temperature enters one end of the unit and passes through a finned tube air heater where its temperature is adjusted to the desired value by allowing steam or water or a mixture of both to flow on the inside of the tubes. After leaving the air heater, the air passes through a short section of duct of reduced cross section where it is thoroughly mixed and its temperature measured by a high-velocity iron-constantan thermocouple. As shown in the drawing, the air in the mixing chamber is protected from direct radiation from the light source by partitioning the duct and drawing a stream of air through the upper half; this insulating air after being partially heated by radiation from the source is discarded into the room. The air leaving the mixing chamber enters the heat collection chamber through a perforated baffle.

The heat collection chamber consists of a number of glass plates defining a plurality of zones or passages through which the air is passed. These plates are arranged in staggered relation to each other and lie in a horizontal position. A section of the top surface of each plate adjacent to the trailing edge was made opaque by the use of black paint, and the corresponding bottom surface was coated with a layer of aluminum foil. This assembly is sealed from above by a continuous sheet of glass laid into a cover support. The construction is similar to that of the outdoor unit described in Section III below, and further details are presented in that section.

An artificial source of solar energy in the form of a bank of 84 - 1500 watt electric tungsten lamps is situated 10 feet above the cover glass of the heat collection chamber, and interposed between the source and the cover glass is a sheet of double strength window glass which absorbs the excess infrared radiation so that the radiation impinging on the collection plates more nearly approximates that from the sun.

The heat collection chamber described makes up a length of 6 feet of the total length of the indoor unit. Methods of measuring temperatures in this chamber make use of high-velocity iron-constantan thermocouples and chromel-constantan foil surface thermocouples. Their positions are described below.

The heated air leaving the heat collection chamber passes to a tapered exit section which is well insulated against radiant heat. At the apex or outlet of the tapered section is a shielded iron-constantan thermocouple for the measurement of the exit air temperatures. The air is then routed through a calibrated rotameter where the flow rate is measured; an iron-constantan thermocouple is also installed at this point. The air leaving the rotameter is then exhausted outdoors through a forge type blower which induces the necessary air flow through the system. All temperature measurements are recorded on a Brown 12-point recording potentiometric pyrometer.

B. Operation

To made a run on the indoor unit, the independent variables are chosen and the unit set in operation. These variables are: Spacing between plates, length of plate overlap, entrance air temperature, and air velocity. When equilibrium is reached, as shown by the attainment of constant temperatures, the run is discontinued, and a new set of independent variables is chosen. At equilibrium the following readings are taken:

1. Air velocity, CFM (760 mm. - 70°F.) (Fischer and Porter Rotameter).

2. Entrance air temperature.

3. Exit air temperature.

4. Temperature of air leaving black surfaces of two plates near the center of the stack.

5. Temperature of air leaving black surfaces of bottom and second plates.

6. Temperature of air passing through the rotameter.

7. Surface temperature in center of blackened portion of central plate in stack. (Bottom plate temperature)

8. Surface temperature three inches behind leading edge of central plate in stack. (Top plate temperature)

9. Cover plate temperature, under side.

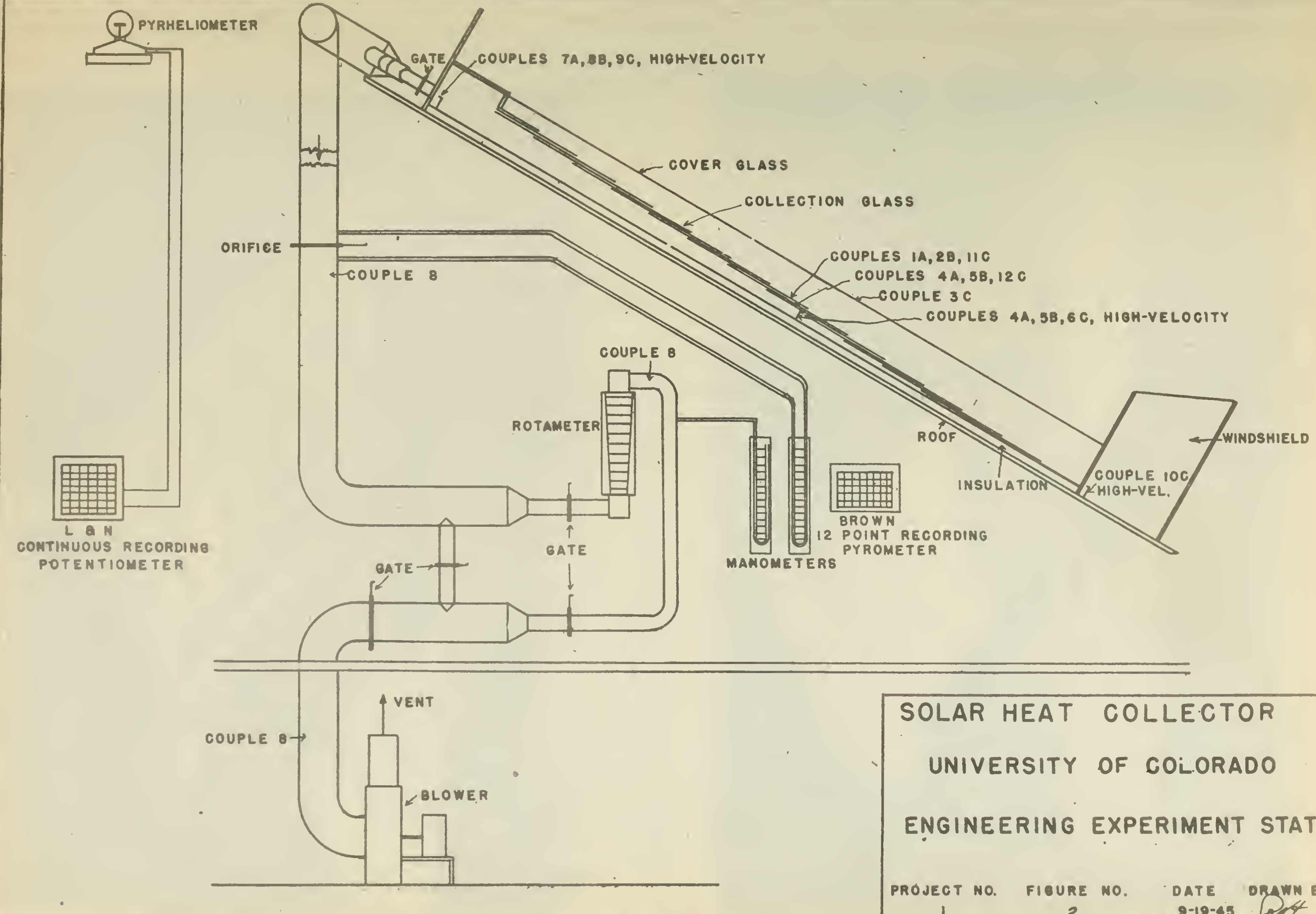
From these data it is possible to calculate the efficiency of heat recovery if the heat input from the lamp bank is known. This input was obtained by placing an Epply pyrheliometer on the unit and measuring the heat input in calories per square centimeter per second.

The data gathered as a result of the above procedure were correlated by a series of graphs as shown in Progress Reports IV and V. (14,15) The important conclusions of the tests on the indoor unit were (1) the general principle of the heat collector as set forth by K. W. Miller was shown to be workable, (2) the optimum plate spacing was found to be 1/4 inch, (3) the optimum plate overlap was found to be 2/3. The plate length factor was not investigated thoroughly enough to draw any definite conclusions, but it is believed that a variation in plate length would not materially affect the operating characteristics of the unit, all other factors being constant.

III. The Outdoor Laboratory Unit

A. Construction

The outdoor unit was constructed on the roof of the Chemical Engineering Laboratory just over the balcony on which the indoor unit was built. The construction will be described with reference to Figure No.2 and the accompanying photograph. The dimensions of the collector are 15 foot wide and 18-1/2 foot long, the exposed portion making up 15 feet of the overall length. Components of the collector are; a floor laid on a double layer of balsam-wool insulation, two supported wood sides adjustable in height, a windshield at the lower end to damper the inlet air from sudden drafts, a board at the upper end with five exit air ports, and five cover glass frames running the width of the unit. Each cover glass support consists of a rectangular framework of 2" x 4" lumber divided into three section



SOLAR HEAT COLLECTOR

UNIVERSITY OF COLORADO

ENGINEERING EXPERIMENT STAT.

PROJECT NO.	FIGURE NO.	DATE	DRAWN BY
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Plate V

The top edges of the timbers are grooved to hold the plates of glass, and each plate is puttied into place. The cover glass supports are doweled together in position and sealed with adhesive tape.

The glass collector plates used in the unit are 36 inches wide, 48 inches long, and 1/8 inch thick; they were supplied by the Jeanette plant of the American Window Glass Company. One third (sixteen inches) of each collector plate was painted with black paint on one side and coated with aluminum foil .00025 inch thick on the other. The black paint contained a solvent made of 60 per cent by volume of turpentine and 40 percent bronzing fluid, and 15 grams of lampblack pigment in 100 cc. of solvent. This paint was compounded after making a series of tests which are explained in Appendix A.

The glass collector plates are supported in the following manner: the bottom black plate is laid flush on the floor at the lower end of the unit and a strip of wood 1-3/4 inches wide and the thickness of the glass is placed on the floor directly behind the glass and extending to the top of the unit where it is secured by a 1 inch spindle. These strips are placed so that their longitudinal center is in line with the junction between two adjacent glass plates. On top of these strips are placed similar strips the thickness of which represents the plate spacing desired. These strips extend from the spindle at the top of the unit to one-third the distance from the bottom end of the bottom plate to the top end of the plate for an overlap of 2/3. At the end of these strips are secured small protruding strips which form a ledge to support the next glass plate. This method of stacking is carried out until all the collector plates are in place. To prevent air flow over the top of the top plate thus short-circuiting the space between the plates, a piece of sheet rubber material compounded from heat resisting neoprene was fastened to the cover plate support and the trailing edge of the top plate. This method of stacking the glass plates may be visualized by inspecting the drawing and photograph.

The air to be heated is drawn into the windshield through the top and ends and through the perforated end board into the collector proper. After passing through the heat collector plates the air is drawn through the exit ports, which are equipped with adjustable dampers, and through a ten inch galvanized iron duct through the roof and into the building. After passing through measuring devices described below, the air passes to the main floor of the laboratory and into a large blower (American Blower, type 250 E) from which the air is exhausted either outdoors or directly into the laboratory when heat is required.

Three types of thermocouples are installed in the unit to measure the necessary temperatures. The surface temperatures are measured with surface thermocouples made of chromel-constantan bright metal foil, 0.0009 inch in thickness and 1/16 inch in width held firmly against the surface by a microscope cover glass glued down with shellac. Chromel and constantan No. 36 B. and S. wires are

soldered to the ends of the foil and pass to the cold junction ice bath. Air temperatures are measured with iron-constantan thermocouples, No. 36 B. and S. Gauge, mounted in 1/8 inch i. d. x 1/2 inch o. d. bakelite cylinders and provided with copper tips having a 3/64 inch opening, through which a small stream of hot air leaving the plates is rapidly drawn by a water aspirator. This high-velocity thermocouple is used to reduce radiation errors. Air temperatures at the rotameter and blower are measured with bare iron-constantan thermocouples inserted in the duct. All thermocouples are connected to a Brown 12-point recording potentiometric pyrometer which is calibrated for iron-constantan thermocouples, and is cold junction compensating. To evaluate the temperatures of the surface couples which are equipped with an ice bath cold junction, it is necessary to subtract the temperature of the potentiometer box from the recorded temperature and convert to chromel-constantan temperature by means of the chart shown in the appendix (Figure 32), which was constructed from tables furnished by The Leeds and Northrup Company.

B. Operation

During a day's run of the solar heat collecting unit, the following temperatures are taken in the manner described above and automatically printed on the potentiometer chart with the numbers given in parentheses: entrance air (10); air leaving the trailing edge of the center plate in the west section or section A (4); air leaving the trailing edge of the center plate in the section adjacent to the west section or section B (5); the air leaving the trailing edge of the center plate of the center section or section C (6); the exit air in sections A, B, and C, (7, 8, and 9); air temperature at the orifice, rotameter, or blower (8); center black plate temperature in sections A, B, and C, (4, 5, 12); center clear plate temperatures in sections A, B, and C, (1, 2, 11); cover glass temperature at the geometric center of the unit (3); cold junction temperature at the recorder (3).

The rate of air flow through the unit is measured in two separate ways. For flow rates up to 150 CFM a Fischer and Porter rotameter equipped with a guided plumb-bob float is used for measurement. The rotameter was calibrated at the factory to read in CFM at 760 mm. and 70°F. when measured at 630 mm. and 70°F. For flow rates above 150 CFM a thin plate 3½ inch orifice installed in the ten inch duct was used. The orifice was calibrated against the rotameter up to flow rates of 250 CFM and the curve extrapolated to 400 CFM. An alignment chart was prepared for rapid calculation of the flow rates. In order that flow rates could be calculated, static pressures were measured by water manometers, and the pressure drop across the orifice was read with a vertical 20 inch water manometer and an inclined water gauge (not shown). In order to make it possible to obtain flow rate measurements over the entire period of a run, a thermocouple was installed in the blower and by consideration of the fact that a constant volumetric rate is handled by the blower, the actual flow rate measured at 760 mm. and 70°F. could be calculated.

The heat input to the solar heat collector was measured by an Eppley 50-junction (23) pyrheliometer which was previously calibrated by the U. S. Weather Bureau. The readings were taken continuously on a Leeds and Northrup continuous recording potentiometer and integrated hourly with a polar planimeter. The instrument was installed on the peak of the roof in a horizontal position as required by specifications of the U. S. Weather Bureau.

Before the unit was set in operation, measurements were made to determine if the flow through each section was equal. This was approximated by determining the static pressure at the exit ports of the five sections. The dampers were then adjusted to obtain equal static pressure.

The unit was first stacked with the optimum plate spacing and overlap as determined by tests on the indoor unit (1/4 inch spacing, 2/3 overlap) and a series of 40 runs was made from July 24 to September 23, 1944, with air rates varying from 53 to 369 CFM and with heat input varying according to the position of the sun and prevailing weather. This procedure was then repeated with 1/2 inch spacing, and a series of 28 runs were made from October 7 to October 31, 1944. The unit was then run periodically through the winter months to determine the variation of performance with time of year. At the end of the following summer (1945) a final run was made with the unit in a bad state of disrepair. Approximately 85% of the plates were broken, water had washed a considerable amount of paint off and all of the air seals were broken. Following the runs, all pertinent data were calculated and put in usable form (see Tabulated Results). Finally, important correlations were made and conclusions drawn as to the performance characteristics of such a solar heating unit.

Data that were necessary for a proper heat balance but that were not directly available were gathered from other sources of information. The atmospheric air temperatures were secured from the U. S. Weather Bureau station on the University Campus. Wind velocity figures were obtained from the Valmont plant of the Public Service Company and corrected for the difference between Boulder and Valmont using figures available a few years ago. These data were necessary for the calculation of convection losses from the top of the unit.

IV. The Experimental House Unit

The experimental house unit was constructed on the roof of the home of Dr. G. O. G. Lof at 1719 Mariposa Street, Boulder, Colorado. Figure No. 3 and the accompanying photograph illustrate the construction of the unit.

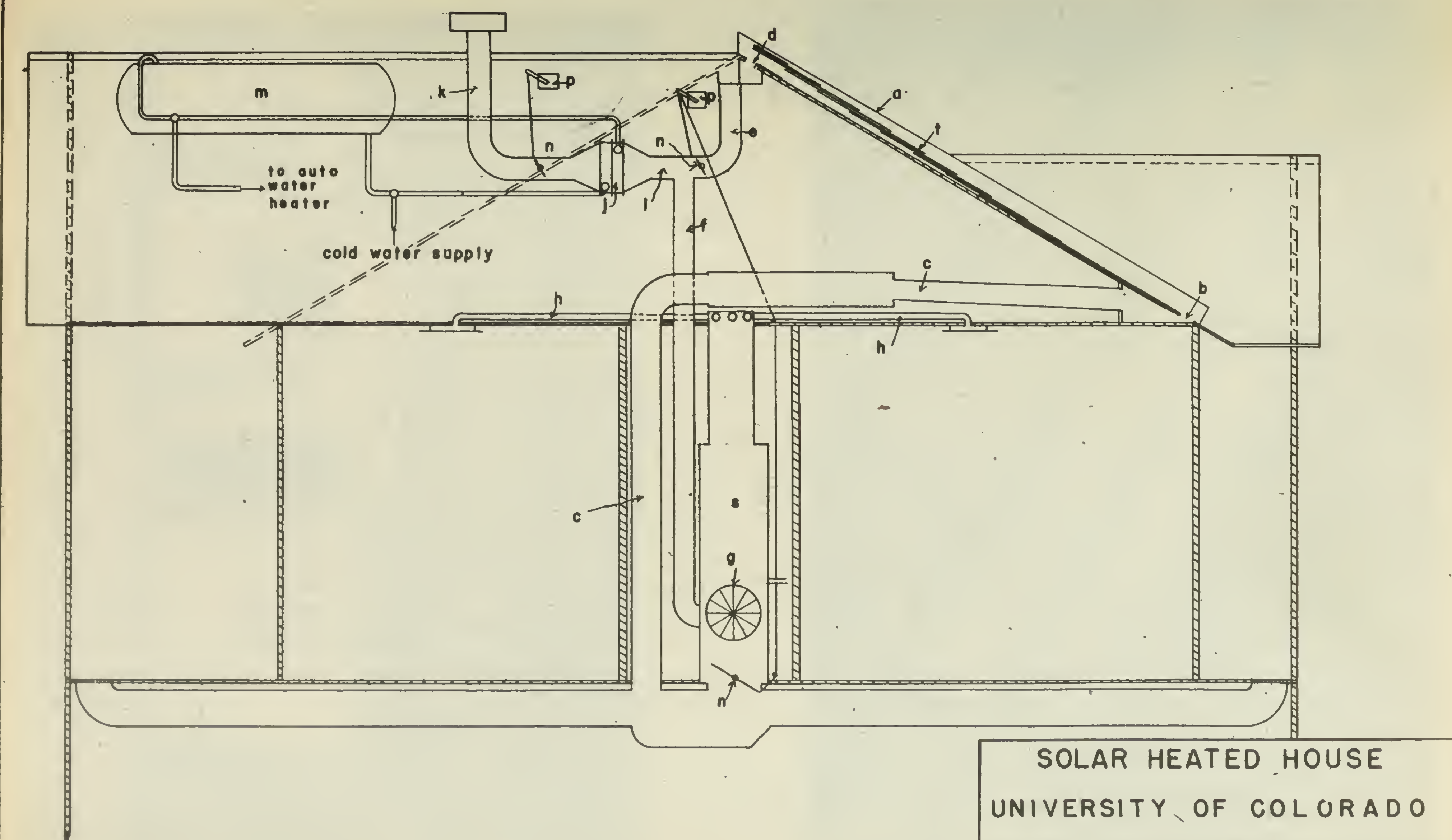
Lying on the slope of the roof are sixteen supports constructed of 2" by 6" lumber supported and separated by iron strips which also act as supports for the cover glasses (a). In the sides of these pieces are cut appropriate slots into which are driven narrow strips of composition board; the slots and strips are positioned so that they will support the glass plates (t) in their proper relation. At the end of each strip is fastened a small wood block which prevents the glass plates from sliding down the roof. The assembly is built directly on top of the shingles of the roof but before the glass is laid, a 1/2 inch layer of celotex insulation is placed over the shingles. The supports are spaced on 32 inch centers and the glass plates are 29 inches wide by 36 inches long. The glass plates are placed in the optimum arrangement found in the tests on the outdoor unit which was found to be 1/4" spacing, 2/3 overlap with 12 inches of each plate painted black. The cover glasses are put in place in groves cut in the top side of the supports and sealed with putty; the ends of the cover glasses overlap in the manner of a green-house glazing. A cap strip of sheet metal is placed on the top edge of each 2 by 6 to protect the seal from weathering. A flashing made of sheet metal covers the top and bottom ends of the unit.

The entrance air inlet (b) is cut through the roof at the bottom end of the unit and a duct (c) is built from the house cold air return (r) to these openings in the roof. The heated air is drawn from the top of the unit through similar holes (d) cut in the roof. The duct system employed for the hot air is shown in the drawing and consists of a large duct (e) leading from the hot air outlets of the unit to the center of the attic where it branches into two. One branch (f) leads to the bottom of the furnace (s) and thence to the fan (g) and into the house hot air circulation system (h). The other branch (i) leads through a finned tube water heater (j) and thence to a natural circulation vent (k) through the roof, the top of which is higher than the top of the solar unit. The finned tube heater is used to heat water when the heat is not needed to heat the house. This heated water is stored in a 120 gallon tank (m) installed in the attic as shown in the drawing. This tank is connected directly to the inlet of the automatic hot water heater.

Installed in this duct system are suitable controls in the form of thermostats, dampers (n), damper motors (p), and relays which permit the unit to be operated in the following manner automatically:

Case I. House is cold, solar unit is cold.
The automatic gas furnace heats the house in the normal manner.

Case II. House is cold, solar unit is hot.
The hot air from the unit goes directly to the furnace inlet and is blown by the furnace fan to the house heating duct system; the cold air return is routed back to the bottom of the solar unit.



SOLAR HEATED HOUSE

UNIVERSITY OF COLORADO

ENGINEERING EXPERIMENT STAT.

PROJECT NO.	FIGURE NO.	DATE	DRAWN BY
1	3	11-30-45	RWH

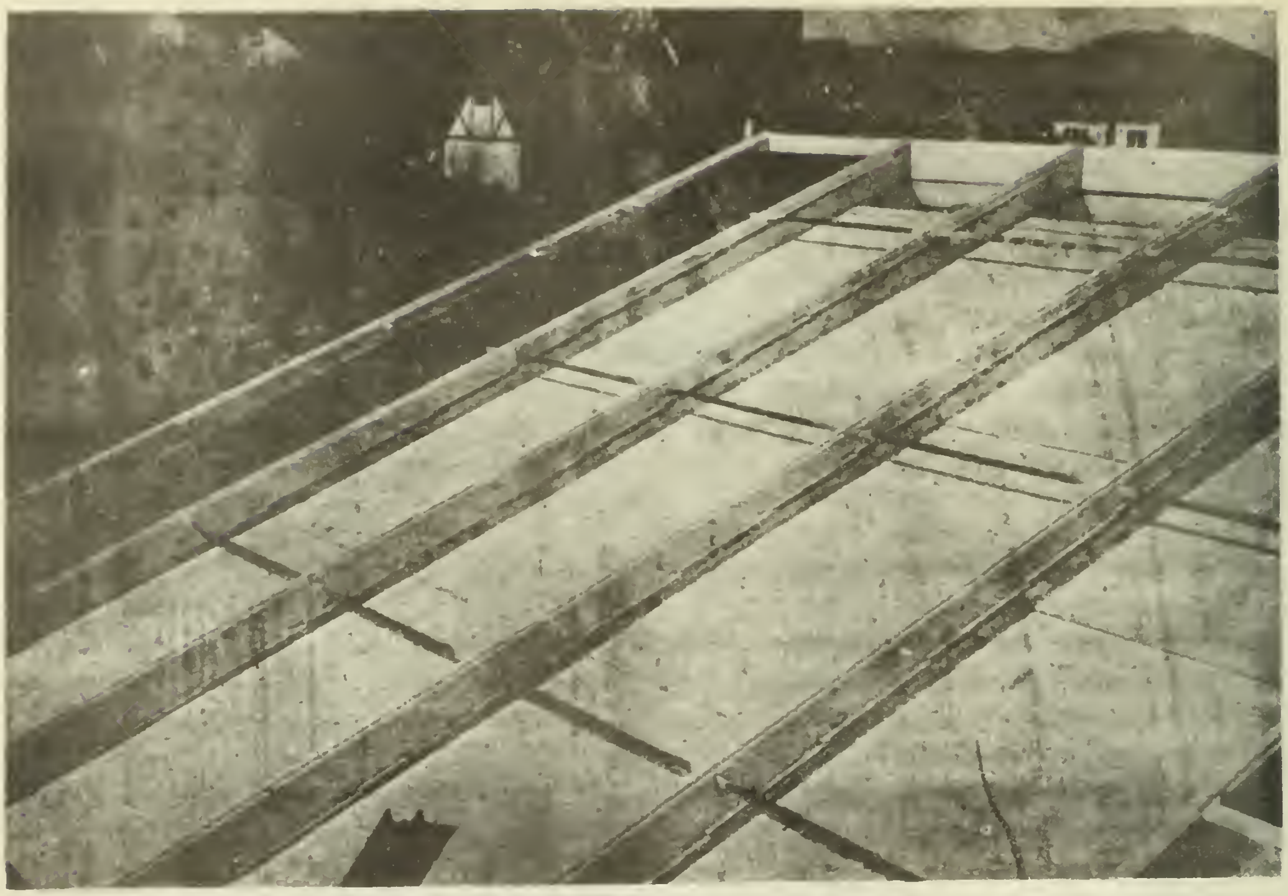


Plate 2

Case III. House is hot, solar unit is cold.
Heating not necessary, therefore dampers are closed and furnace is not operating.

Case IV. House is hot, solar unit is hot.
The hot air from the unit is vented through the natural draft vent after passing through the finned tube water heater.

Before the unit was placed in operation the exit air temperature from each section was set at the same value by adjusting the air flow in the sections with dampers at the outlet (d) of each section. The temperatures were measured by iron-constantan thermocouples.

It was found necessary or desirable to prevent breakage during the hail season by covering the solar heat collecting unit with a suitable screen. A galvanized iron screen with 1/4 inch openings was used. This screen was supported on strips of wood 2 x 2 inches fastened to the top of each 2 x 6.

It was found that no protection from snow was necessary since the tilt of the roof and the smoothness of the glass allowed the snow to slide readily from the unit.

By preliminary calculations and observations it is found that the installation of the unit of 463 sq. ft. on the roof of this house will furnish approximately 33% of the heating load required by the house for an entire winter if no storage facilities are available. Further calculations were made to show the advantage of a suitable storage unit for the storage of heat for one, two, and three day periods. These calculations were based on the heat required during the previous year per degree-day of heating load, which was determined from gas meter readings and degree-day figures procured from The Public Service Company.

During the heating season, the gas meters in the test house and in the identical adjacent house were read each month and compared in order to determine the fuel saving effected by the solar unit. The gas consumptions in the test house before and after installation of the solar unit were also compared on a degree-day basis.

V PRIMARY RESULTS

TABLE OF RESULTS I

Overlap 2/3" Plate Length 48" Collector Area = 209 ft²

Run No. (Outdoor Unit)	1	2	7	8	9	10	11	12	13	15	16	17	18	19	20	21
Date 1944	7/24	7/25	8/11	8/12	8/13	8/14	8/15	8/16	8/17	8/19	8/20	8/21	8/22	8/23	8/24	8/25
Spacing, inches	1/4	1/4	1/4	1/4	1/4	1/4	1/4	1/4	1/4	1/4	1/4	1/4	1/4	1/4	1/4	1/4
Sunrise, A.M.	4:49	4:50	5:06	5:07	5:08	5:09	5:10	5:11	5:12	5:15	5:16	5:17	5:18	5:20	5:21	5:22
Sunset, P.M.	7:10	7:09	6:53	6:57	6:57	6:50	6:50	6:50	6:46	6:44	6:42	6:42	6:41	6:39	6:38	6:37
First sun on collector, A.M.	5:41	5:41	5:45	5:45	5:46	5:46	5:46	5:47	5:47	5:47	5:48	5:48	5:48	5:49	5:49	5:49
Last sun on collector, P.M.	6:19	6:19	6:15	6:14	6:14	6:14	6:13	6:13	6:13	6:12	6:12	6:12	6:11	6:11	6:11	6:10
Cloud loss % *	46	7	0	0	0	34	33	23	0	33	23	12	24	59	58	16
Wind velocity m.p.h.			3													
Bulk air temp., °F			76													
Av. entrance air temp., °F	79	80	86	85	81	79	71	76	73	79	76	77	78	68	68	77
Av. air temp. rise, °F	34	43	68	80	71	49	55	59	73	46	54	46	39	22	21	41
Av. exit air temp., °F	113	123	154	165	152	128	126	135	146	125	130	123	117	90	89	118
Max. exit air temp., °F	133	184	224	229	231	182	204	224	223	200	210	193	188	151	171	200
Av. cover temp., °F			102				56	63	67	57			54			58
Av. black plate temp., °F			185		171				180							
Air rate, CFM (760mm-70°F)	335	360	85	93	94	61	56	56	32	105	100	212	207	216	220	212
Heat recovered BTU/ft²	404	719	451	523	522	224	259	276	201	380	457	808	722	413	400	739
Gross heat input BTU/ft²	1111	1895	2524	2589	2561	1676	1706	1869	2425	1626	1865	2130	1914	1071	1093	2195
Reflection loss BTU/ft²	316	462	666	664	679	469	425	463	631	416	467	570	489	300	306	547
Net heat input BTU/ft²	795	1433	1858	1925	1882	1207	1281	1406	1794	1210	1398	1560	1425	771	787	1648
Gross efficiency, %	36.4	38.0	17.8	20.2	20.4	13.3	15.2	14.8	8.3	23.3	24.5	37.9	37.7	38.5	36.5	33.7
Net efficiency, %	50.8	50.2	24.3	27.2	27.8	18.5	20.2	19.7	11.2	31.4	32.6	51.7	50.7	53.5	50.8	44.9

Run No. (Outdoor Unit)	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38	39
Date 1944	8/26	8/27	8/28	8/29	8/30	8/31	9/1	9/2	9/5	9/6	9/7	9/11	9/12	9/13	9/14	9/15	9/16	9/17
Spacing, inches	1/4	1/4	1/4	1/4	1/4	1/4	1/4	1/4	1/4	1/4	1/4	1/4	1/4	1/4	1/4	1/4	1/4	1/4
Sunrise, A.M.	5:23	5:25	5:26	5:27	5:28	5:29	5:31	5:32	5:36	5:37	5:38	5:43	5:44	5:45	5:47	5:48	5:50	5:51
Sunset, P.M.	6:36	6:35	6:33	6:33	6:31	6:30	6:29	6:27	6:24	6:22	6:21	6:16	6:15	6:14	6:12	6:11	6:10	6:08
First sun on collector, A.M.	5:50	5:50	5:51	5:51	5:51	5:52	5:52	5:52	5:53	5:54	5:54	5:55	5:56	5:56	5:56	5:57	5:57	5:58
Last sun on collector, P.M.	6:10	6:10	6:09	6:09	6:08	6:08	6:08	6:07	6:06	6:06	6:06	6:04	6:04	6:04	6:03	6:03	6:03	6:02
Cloud loss % *	59	0	12	72	0	3	39	9	8	0	5	0	0	14	40	30	0	0
Wind velocity m.p.h.															4		2	2
Bulk air temp., °F															69		62	69
Av. entrance air temp., °F	65	74	82	57	69	70	75	74	72	77	81	63	66	74	72	66	70	77
Av. air temp. rise, °F	24	37	35	13	30	40	19	27	62	69	75	45	51	70	42	38	42	28
Av. exit air temp., °F	89	111	121	70	99	110	94	101	134	146	156	108	117	144	114	104	112	105
Max. exit air temp., °F	117	176	182	92	159	166	120	148	215	221	221	168	159	195	172	126	186	166
Av. cover temp., °F						77			63	67	70	52		107	79	86	114	82
Av. black plate temp., °F							98								195		137	129
Air rate, CFM (760mm-70°F)	214	290	200	336	369	332	332	314	95	94	53	274	210	56	56	222	225	367
Heat recovered BTU/ft²	371	908	688	381	942	932	526	695	513	508	284	941	864	263	194	553	797	884
Gross heat input BTU/ft²	1074	2620	2316	737	2490	2416	1529	2277	2228	2420	2305	2427	2421	1740	1452	1698	2424	2355
Reflection loss BTU/ft²	301	674	605	206	640	611	436	584	574	619	600	626	620	416	359	436	621	600
Net heat input BTU/ft²	773	1946	1711	531	1850	1805	1093	1693	1654	1801	1705	1801	1801	1324	1053	1262	1803	1755
Gross efficiency, %	34.5	34.6	29.7	51.7	37.9	38.6	34.4	30.5	23.0	21.0	12.3	38.8	35.7	15.1	13.4	32.6	32.9	37.4
Net efficiency, %	48.0	46.7	40.2	71.8	50.9	51.6	48.0	41.0	31.0	28.2	16.7	52.2	48.0	19.9	18.4	43.8	44.2	50.4

Heat balance BTU/ft²:	Run 7	Run 36	Run 39
Reflection loss	666	399	600
Convection loss	900	457	407
Conduction loss	306	265	158
Re-radiation loss	477	176	235
Unaccounted for losses	-277	-40	73
Total losses	2072	1257	1473
Percentage losses:			
Reflection	32.1	31.8	40.7
Convection	43.4	36.3	27.6
Conduction	14.7	21.1	10.7
Re-radiation	23.0	14.0	16.0
Unaccounted for losses	-13.2	-3.2	5.0

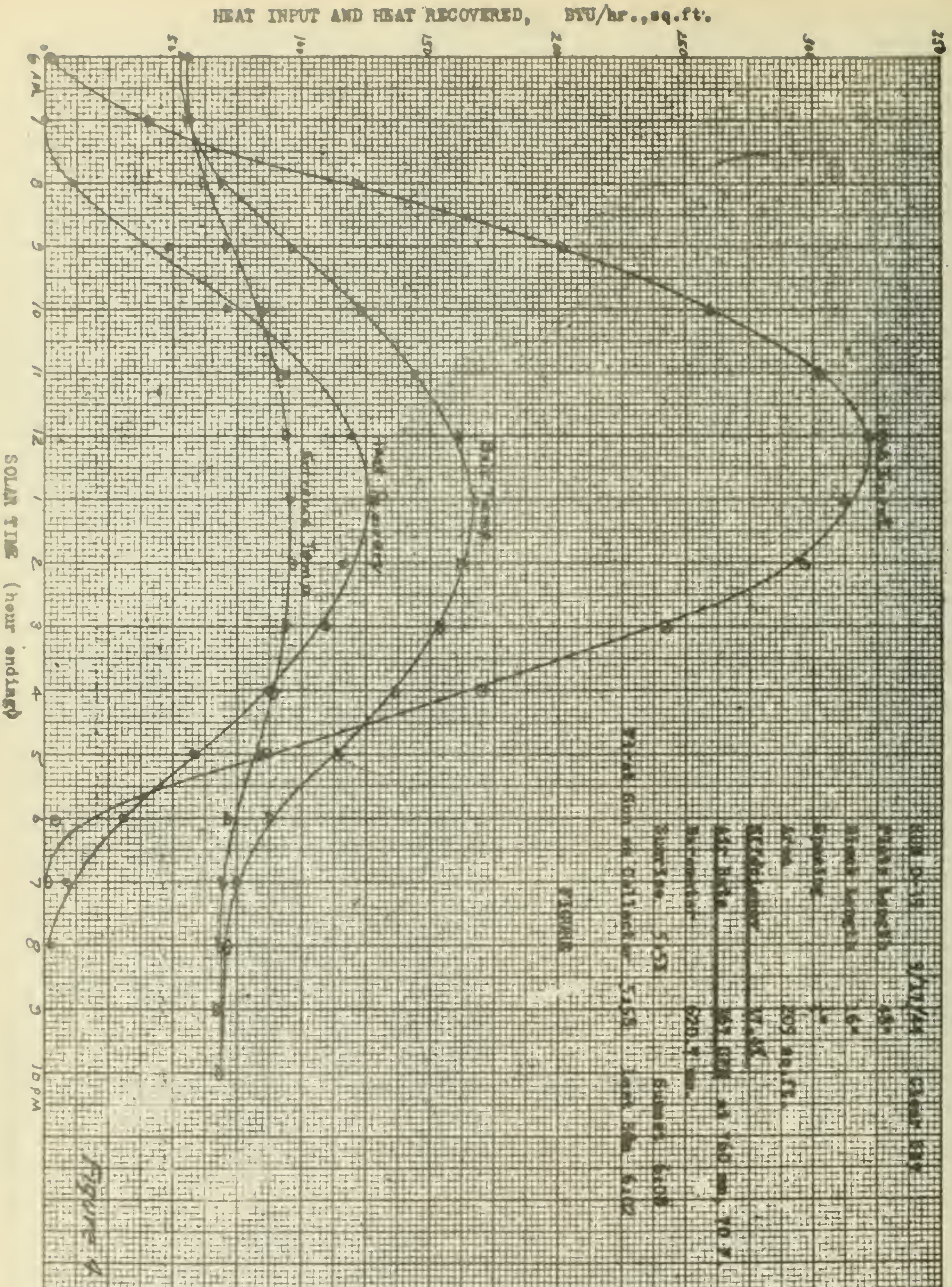
Run No. (Outdoor Unit)	40	41	42	43	44	45	46	47	48	49	50	51	52	54	55	56
Date	9/18	9/19	9/20	9/21	9/22	9/23	10/7	10/8	10/9	10/11	10/12	10/13	10/14	10/17	10/19	10/20
Spacing, Inches	1/4	1/4	1/4	1/4	1/4	1/4	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2
Sunrise, A.M.	5:52	5:53	5:55	5:56	5:58	6:00	6:18	6:19	6:21	6:23	6:25	6:26	6:27	6:31	6:33	6:35
Sunset, P.M.	6:07	6:06	6:04	6:03	6:02	6:00	5:42	5:41	5:40	5:36	5:36	5:34	5:33	5:30	5:27	5:26
First sun on collector	5:58	5:58	5:59	5:59	5:59	6:00	6:18	6:19	6:21	6:23	6:25	6:26	6:27	6:31	6:33	6:35
Last sun on collector	6:02	6:02	6:01	6:01	6:00	6:00	5:42	5:41	5:40	5:36	5:36	5:34	5:33	5:30	5:27	5:26
Cloud loss %	5	7	0	12	8	11	0	19	24	8	12	16	44	0	10	2
Wind velocity m.p.h.	3	2	5			3	2	2	3	2	3	3	4	7	2	3
Bulk air temp., °F			67			57				50	61			57	50	58
Av. entrance air temp, °F	60	63	74	76	78	69	66	70	74	62	74	75	64	61	62	74
Av. air temp. rise, °F	36	30	30	64	64	64	30	28	26	34	48	49	30	63	51	39
Av. exit air temp., °F	96	93	104	140	142	133	96	98	100	96	122	124	94	124	113	113
Max. exit air temp., °F	162	151	169	213	210	207	143	142	146	148	195	192	125	188	183	164
Av. cover temp., °F	66	72	78	124	122	85	66			62	80			76	70	71
Av. black plate temp., °F	117	131	129			155	117			115	176			156	138	129
Air rate, CFM (760-70)	280	333	333	57	57	58	354	272	271	269	90	34	33	33	57	218
Heat recovered BTU/ft²	852	834	843	283	278	319	756	556	499	652	358	142	77	157	236	551
Gross heat input BTU/ft²	2240	2257	2430	2148	2242	2153	2271	1884	1780	2135	2056	1742	1161	2202	1999	2072
Reflection loss BTU/ft²	562	578	625	545	569	527	582	496	483	544	525	441	323	574	513	530
Net heat input BTU/ft²	1678	1679	1805	1603	16[illegible]	1626	1689	1388	1297	1591	1531	1301	838	1628	1486	1542
Gross efficiency, %	38.0	37.0	34.7	13.2	12.4	14.8	33.3	29.4	28.1	30.6	17.4	8.1	6.6	7.1	11.8	26.6
Net efficiency, %	50.8	49.7	46.7	17.6	16.6	19.6	44.8	40.3	38.6	41.0	23.3	10.9	9.2	9.7	15.8	35.7

Run No. (Outdoor Unit)	57	58	59	60	61	62	63	64	65	66	67	68	69	70	71	72	73	74
Date	10/21	10/22	10/23	10/24	10/25	10/26	10/27	10/28	10/29	10/30	10/31	11/17	11/21	12/19	12/20	12/30	2/23*	5/3*
Spacing, Inches	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2
Sunrise, A.M.	6:36	6:37	6:38	6:39	6:41	6:42	6:43	6:45	6:46	6:47	6:49	7:07	7:11	7:26	7:26	7:24	6:35	5:06
Sunset, P.M.	5:29	5:29	5:22	5:20	5:19	5:18	5:16	5:15	5:14	5:13	5:12	4:53	4:50	4:34	4:34	4:36	5:26	6:53
First sun on collector	6:36	6:37	6:38	6:39	6:41	6:42	6:43	6:45	6:46	6:47	6:49	7:07	7:11	7:26	7:26	7:21	6:35	5:45
Last sun on collector	5:29	5:25	5:27	5:20	5:19	5:18	5:16	5:15	5:14	5:13	5:12	4:53	4:50	4:34	4:34	4:36	5:36	6:15
Cloud loss %	4	10	5	4	0	0	0	7	17	40	15	11	0	12	26	0	48	0
Wind velocity m.p.h.	2	7	2	2	2	4	2	2	2	2	3		2	3	6	5	2	3
Bulk air temp., °F																		
Av. entrance air temp, °F	70	67	78	77	70	67	67	65	67	62	68	42	51	44	47	[illegible]	40	58
Av. air temp. rise, °F	39	24	26	56	30	26	30	29	28	17	26	24	33	30	25	38	[illegible]	56
Av. exit air temp., °F	109	51	104	133	106	93	97	98	95	75	54	66	84	74	72	72	64	114
Max. exit air temp., °F	150	137	144	196	150	148	147	144	146	111	139	105	121	116	108	108	76	173
Av. cover temp., °F			72															
Av. black plate temp., °F			121															136
Air rate, CFM (760-70)	217	344	336	32	290	250	294	255	296	255	290	256	298	282	286	275	236	274
Heat recovered BTU/ft²	473	449	519	120	525	504	521	520	464	272	422	430	504	382	315	544	345	1084
Gross heat input BTU/ft²	2126	1876	2021	1932	1978	1514	2096	1547	1735	1235	1650	1521	1717	1566	1316	[illegible]	1320	2648
Reflection loss BTU/ft²	520	453	517	489	489	480	492	484	445	364	451	355	416	439	368	454	370	655
Net heat input BTU/ft²	1606	1423	1504	1443	1489	1434	1604	1463	1290	875	1235	1162	1299	1125	948	1330	950	1953
Gross efficiency, %	22.2	24.0	25.7	6.2	26.6	26.3	24.9	26.7	26.7	21.9	24.5	28.3	25.3	24.4	24.0	30.5	26.4	41.0
Net efficiency, %	25.5	31.5	34.5	8.3	35.2	35.1	32.4	35.5	36.0	31.1	34.0	37.0	38.8	33.8	33.2	41.0	36.8	55.5

Heat balance BTU/ft²	Run 42	Run 45	Run 49	Run 50	Run 54	Run 55	Run 56
Reflection loss	625	527	544	525	575	513	531
Convection loss	551	888	268	577	1235	576	360
Conduction loss	159	278	150	220	258	214	149
Re-radiation loss	213	486	159	363	337	360	201
Unaccounted for losses	40	-342	362	12	-361	99	281
Total losses	1588	1837	1481	1697	2044	1762	1522
Percentage losses:							
Reflection	39.4	28.7	36.7	31.0	28.2	25.1	34.9
Convection	34.7	48.4	18.1	34.0	60.4	32.7	23.7
Conduction	10.0	15.1	10.1	12.9	12.6	12.2	5.8
Re-radiation	13.4	26.4	10.7	21.4	16.5	20.4	13.2
Unaccounted for losses	2.5	-18.6	24.4	0.7	-17.7	5.6	18.4

*1948

* Cloud loss = $\dfrac{\text{Input with clear sky} - \text{actual input}}{\text{Input with clear sky}}$


HEAT INPUT AND HEAT RECOVERED, BTU/hr.,sq.ft.
SOLAR TIME (hour ending)
TEMPERATURE OF AIR ENTERING AND LEAVING, F.
Figure 4

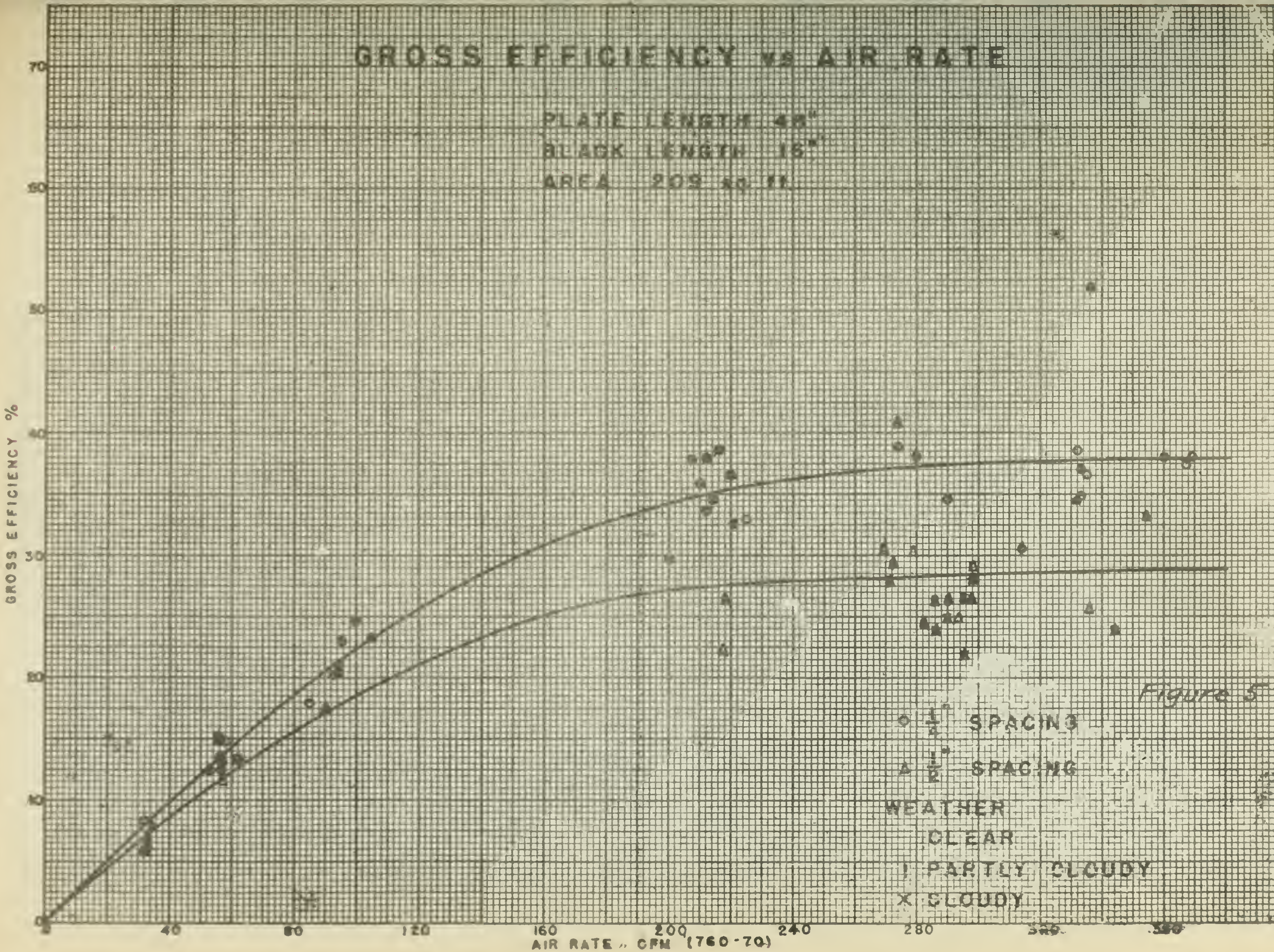
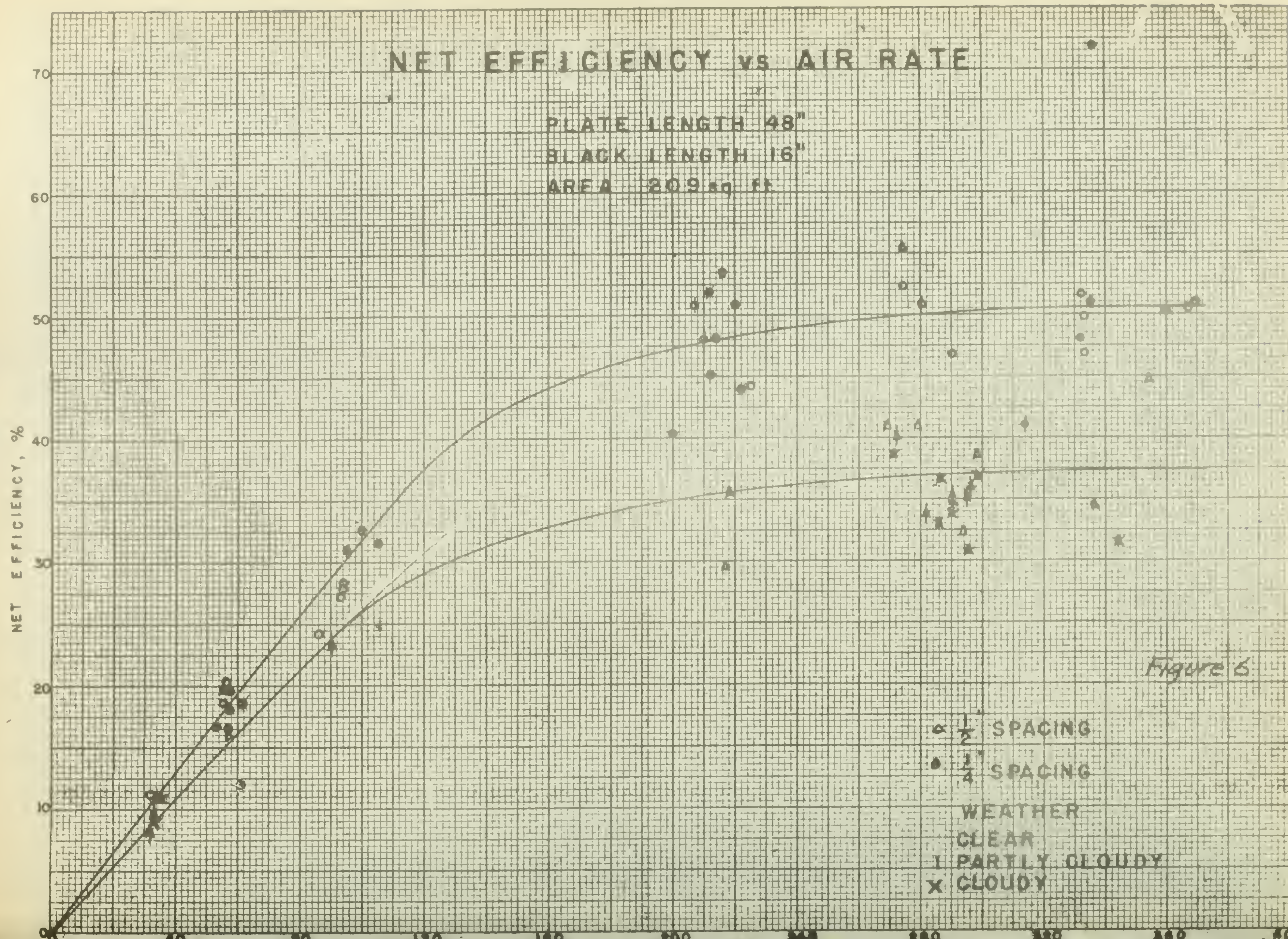

GROSS EFFICIENCY vs AIR RATE
PLATE LENGTH 48"
BLACK LENGTH 16"
AREA 209 sq ft
GROSS EFFICIENCY %
AIR RATE ~ CFM (760-70)
0
40
80
120
160
200
240
280
10
20
30
40
50
60
70
○ ½" SPACING
△ ½" SPACING
WEATHER
CLEAR
| PARTLY CLOUDY
X CLOUDY

Figure 5


NET EFFICIENCY vs AIR RATE
PLATE LENGTH 48"
BLACK LENGTH 16"
AREA 209 sq ft
NET EFFICIENCY, %
70
60
50
40
30
20
10
0
40
80
120
Figure 6
$\frac{1}{2}$" SPACING
$\frac{1}{4}$" SPACING
WEATHER
CLEAR
I PARTLY CLOUDY
X CLOUDY

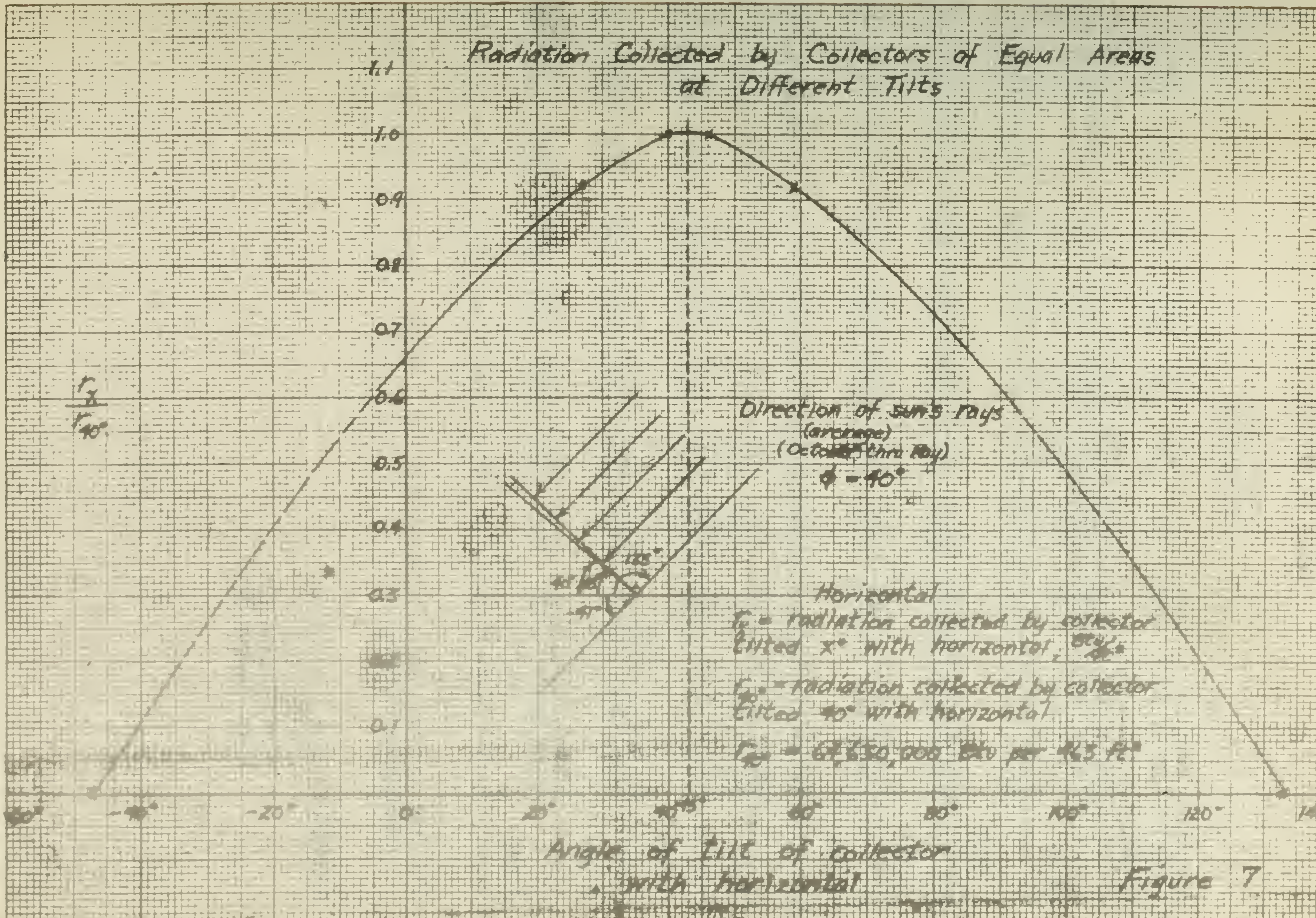

Radiation Collected by Collectors of Equal Areas
at Different Tilts
1.1
1.0
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.1
$\frac{r_x}{r_{40°}}$
Direction of sun's rays
(average)
$\phi = 40°$
Horizontal
r_x = radiation collected by collector tilted x° with horizontal
$r_{40°}$ = radiation collected by collector tilted 40° with horizontal
-20°
0°
20°
40°
60°
80°
100°
120°
140°
Angle of tilt of collector with horizontal

Figure 7

TABLE OF RESULTS II

House Unit
Calculated

Tilt	Storage Capacity (Time)	Heat in BTU x 10^{-3}	Oct. 1944	Nov. 1944	Dec. 1944	Jan. 1945	Feb. 1945	March 1945	April 1945	May 1945	Total
27°		Heat req'd	5416	10393	15036	14657	13471	11403	10402	4229	8500
		Solar rec'd	8400	6703	7110	6154	6087	9193	7963	7983	5964
	None	Solar used	1620	3688	5462	4604	4024	4370	(1)	(1)	2376
		% load cr'd	30.0	35.5	36.3	31.4	30.0	40.2	(1)	(1)	34.
	1 day	Solar used	4530	6198	7110	6154	6029	7816	5813	2918	4656
		% load cr'd	83.6	59.6	47.3	42.0	44.8	68.3	55.8	69.0	54.7
	2 day	Solar used	4723	6793	7110	6154	6087	8358	6627	3621	4947
		% load cr'd	87.2	65.4	47.3	42.0	45.2	73.4	63.6	85.8	58.2
	3 day	Solar used	4723	7036	7110	6154	6087	8382	7536	4053	5108
		% load cr'd	87.2	67.7	47.3	42.0	45.2	78.3	72.3	96.0	60.1
40°		Solar rec'd	9105	7707	8308	7250	7003	9635	8053	7515	6463
	None	Solar used	1623	3804	5647	4760	4204	4416	(1)	(1)	2445
		% load cr'd	30.0	36.6	37.5	32.5	31.2	40.6	(1)	(1)	35.1
	1 day	Solar used	4560	6911	8291	7167	6870	8085	5825	2805	5051
		% load cr'd	84.2	66.5	55.1	49.0	51.0	70.8	56	66.4	59.5
	2 day	Solar used	4726	7413	8308	7250	7003	8614	6685	3582	5358
		% load cr'd	87.2	71.3	55.3	49.5	52.0	75.5	64.3	84.8	63.0
	3 day	Solar used	4726	7727	8308	7250	7003	8679	7644	4017	5535
		% load cr'd	87.2	74.4	55.3	49.5	52.0	76.0	73.5	95.0	65.1

(1) Data not available for April and May. Totals based on requirements through March only.

Collector Area 463 sq. ft.
Overlap 2/3
Plate length 36"
Black length 12"

TABLE OF RESULTS III

PREDICTED SIZE OF HEAT STORAGE UNIT

Storage time	For storage of average load			For storage of maximum		
	Heat st'd BTU	lbs Mat'l	% heat load carried	Heat Stored Btu	Lbs. Mat'l	% heat load cr'd
1 day	300,000	11,500	53.5	400,000	15,400	54.8
2 day	500,000	19,200	56.5	740,000	28,500	58.2
3 day	800,000	30,800		970,000	37,300	60.1

Note: Heat capacity of material 0.2 Btu/#°F
(brickwork, carbon, clay, glass, granite, stone)

Available temperature rise 130°F

Net efficiency of solar unit 45%

Solar unit tilt from horizontal 27°

TABLE OF RESULTS IV

House Unit Performance Data 1945-46

A. Basis: Comparison with identical house

1 Period 1945-46	2 Gas used - Cu ft. (19) Solar House	3 Identical House	4 * Identical +40.5%	5 Gas saved in Solar House	6 % heating load cr'd by solar unit
Sept.10-Oct 12	11,200(19)	9,900	13,900	2,700	19.4
Oct 13-Nov 10	14,400	13,400	18,800	4,400	23.4
Nov 11-Dec 12	23,600	23,900	33,600	10,000	29.8
Dec 13-Jan 12	29,800	26,500	37,200	7,400	19.9
Jan 13-Feb 12	29,100	24,100	33,800	4,700	13.9
Total	108,100	97,800	137,300	29,200	21.2

B. Basis: Degree Day Data

1 Period 1945-46	2 °Days (19)	3 °Days +15.5%	4 Gas Necessary	5 Gas Used	6 Gas for hot water	7 Net gas Used	8 Gas Saved	9 % Heating load Carried by Unit
Sept 10-Oct 12	301	348	8600	11,200	2433	8800	0	0
Oct 13-Nov 10	342	395	9700	14,400	2433	12000	0	0
Nov 11-Dec 12	900	1039	25650	23,600	2433	21200	4450	17.4
Dec 13-Jan 12	1052	1218	30100	29,800	2433	27,400	2700	9.0
Jan 13-Feb 12	1038	1198	29600	29,100	2433	26,700	2900	9.8
Total	3633	4198	103710	108,100	12165	96,100	10050	9.8

C. Operating Data

Area of Unit, sq. ft.	463
Air rate, CFM (760 mm-70°F)	188
Air temperature rise at noon °F	67
Gross efficiency %	10
Net efficiency %	12.8
Predicted heating load carriable %	20.0

* Solar house used 25% more heat than identical house during winter of 1944-45. During winter 1945-46 solar house maintained 5 degrees higher than winter 1944-45. Public Service figures indicate heat requirements will be 15.5% higher for this 5 degree raise, making a total of 40.5% higher than the identical house.

TABLE OF RESULTS V

Comparison of Actual and Theoretical Results

	Experimental (2 clear plates plus cover)		Theoretical	
	Actual	Adjusted	3 plates (2 clear plates)	4 plates (3 clear plates)
Incident Solar Radiation Btu/hr ft^2	329	329	329	329
Top Plate Temperature $^\circ$F	100	100	100	100
Heat Lost Btu/hr ft^2				
convection	57	57	57	57
reradiation	27	27	27	27
reflection	63	63	47	63
unaccounted	24	0	0	0
Heat Recovered Btu/hr ft^2	158	182	198	182
Black Plate Temperature $^\circ$F	225	---	243	290
Air leaving Black Plate $^\circ$F	186	201	224	278
Air Rate (CFM, 760 mm -70°F)	312	---	224	138
Gross Efficiency %	48	55.4	60.2	5[illegible].4

VI SECONDARY RESULTS AND CORRELATIONS

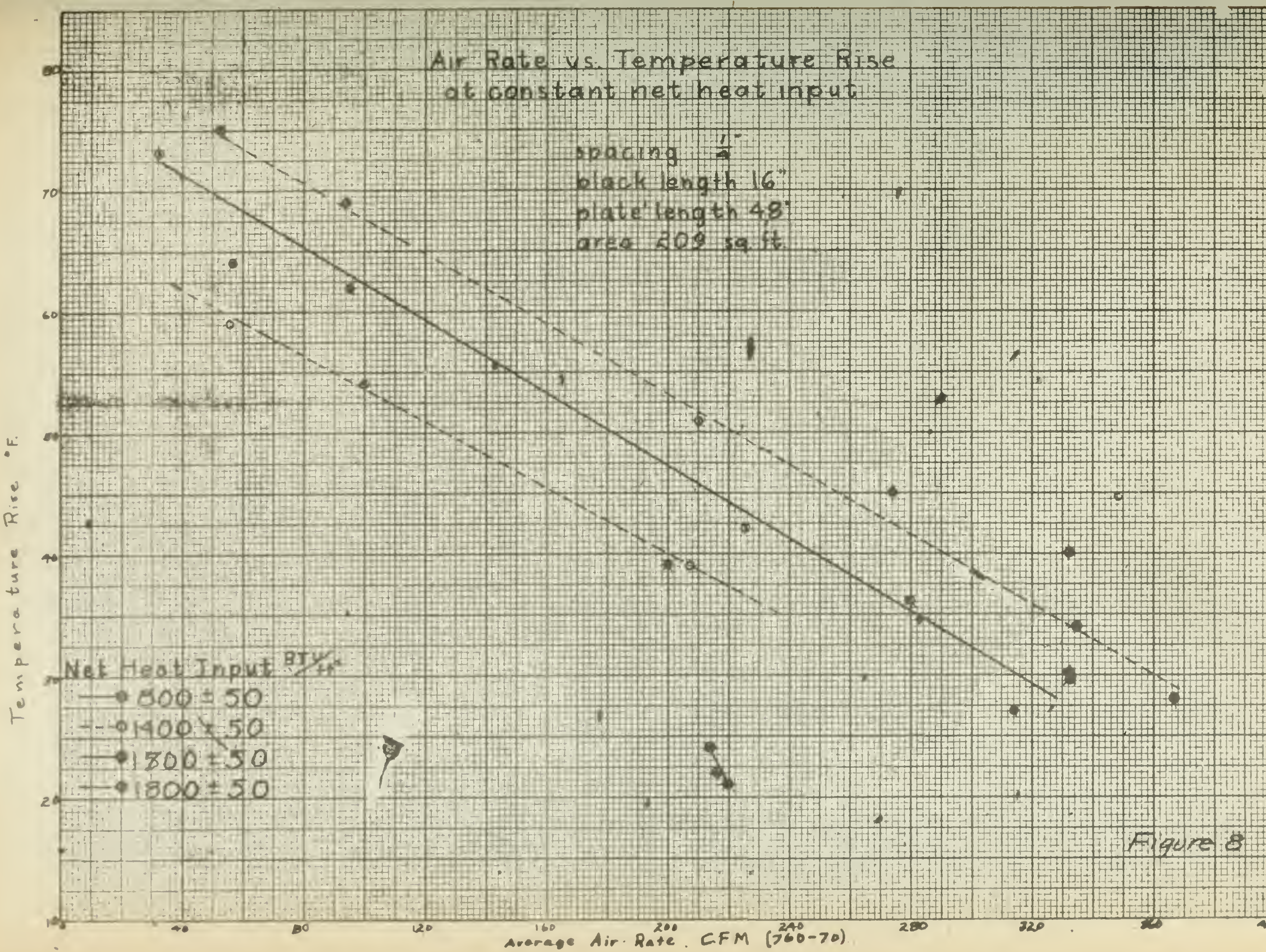
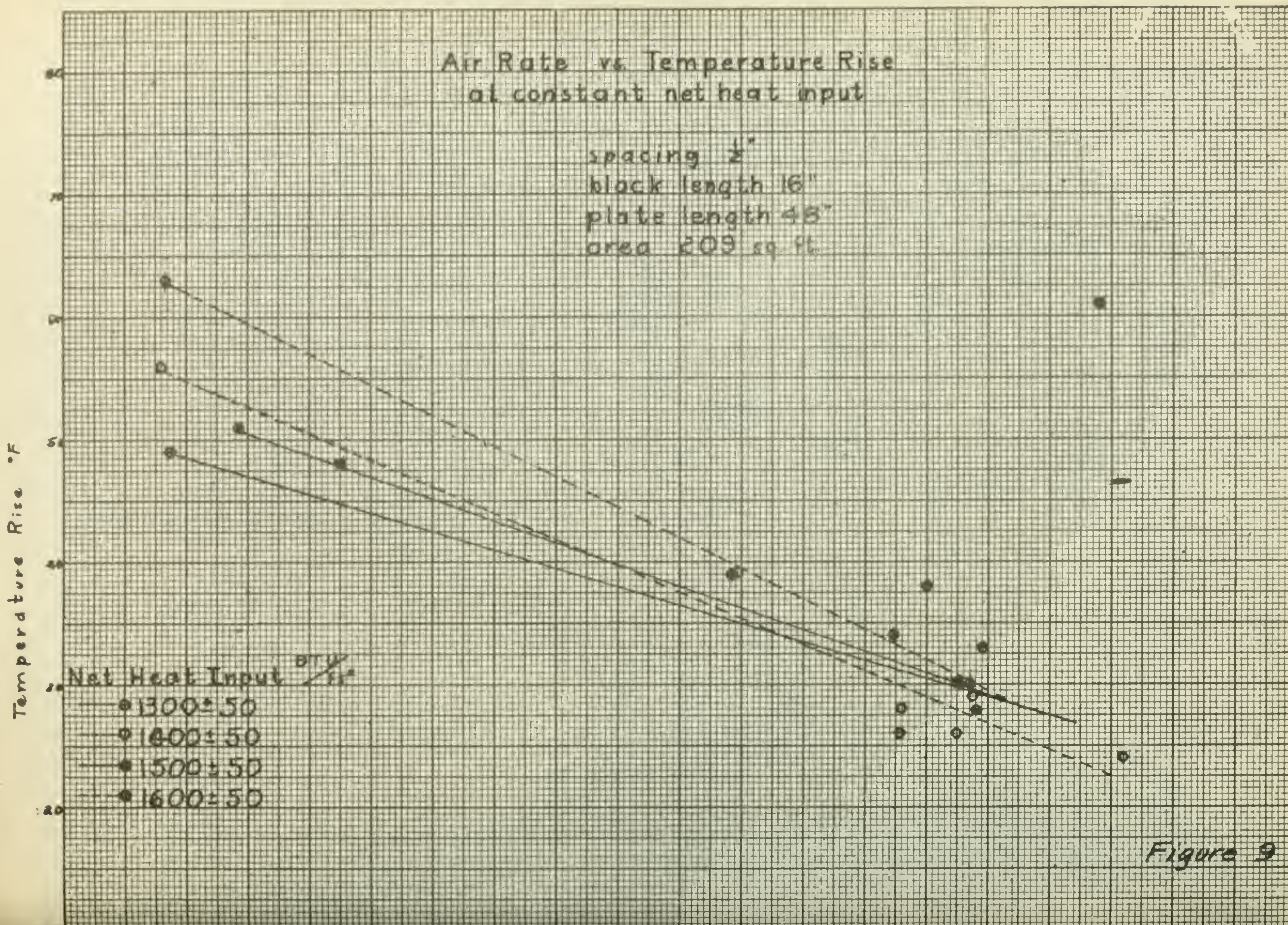

Air Rate vs. Temperature Rise
at constant net heat input
spacing 1/4"
black length 16"
plate length 48"
area 209 sq. ft.
Net Heat Input BTU/hr
800 ± 50
1400 ± 50
1300 ± 50
1800 ± 50
Temperature Rise °F.
Average Air Rate CFM (760-70)
10
20
40
60
70
80
40
80
120
160
200
240
280
320
400

Figure 8


Air Rate vs. Temperature Rise
at constant net heat input
spacing 1/2"
black length 16"
plate length 48"
area 2.09 sq. ft.
Net Heat Input BTU/hr
1300±50
1400±50
1500±50
1600±50
Temperature Rise °F
Figure 9

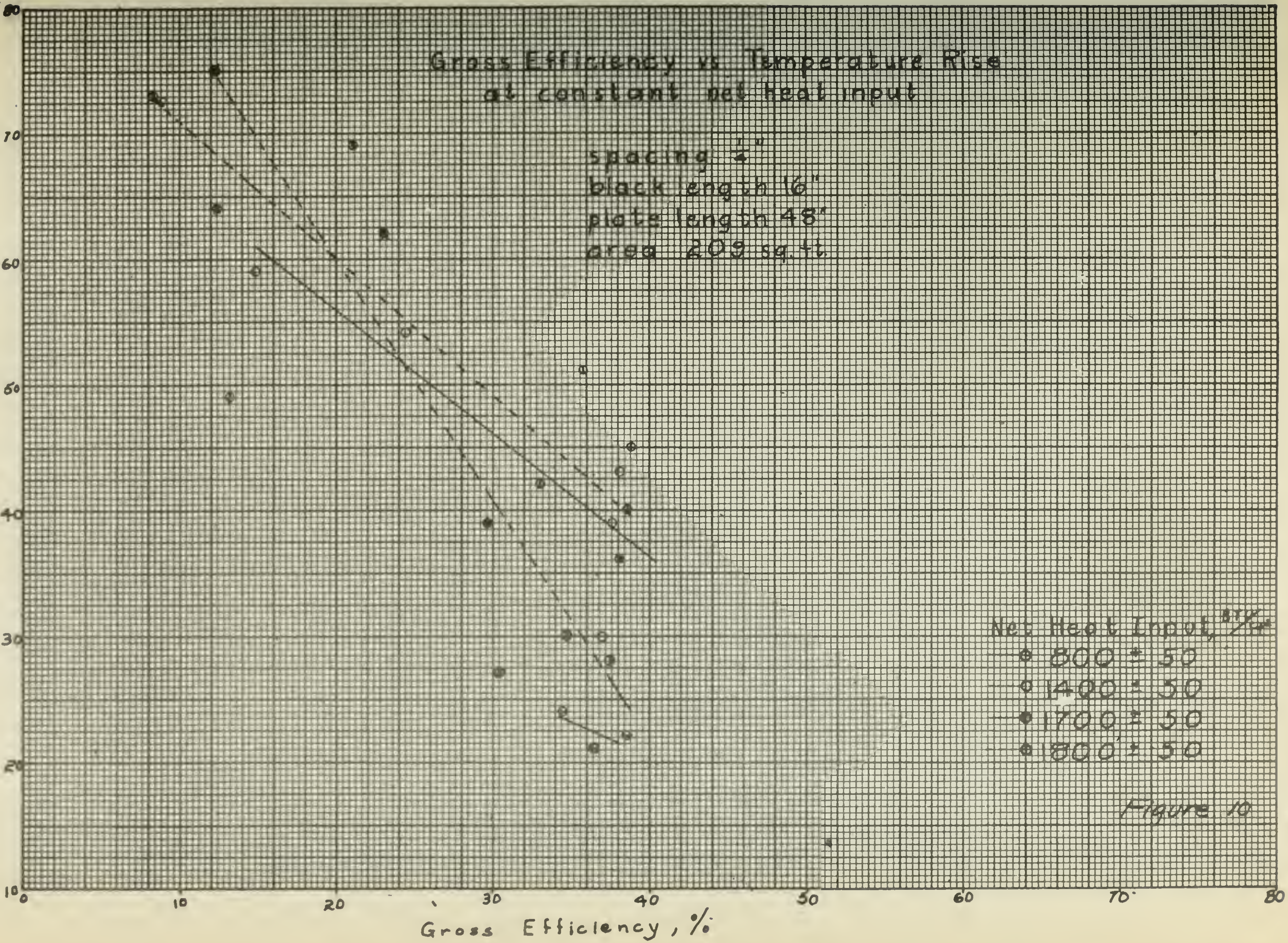
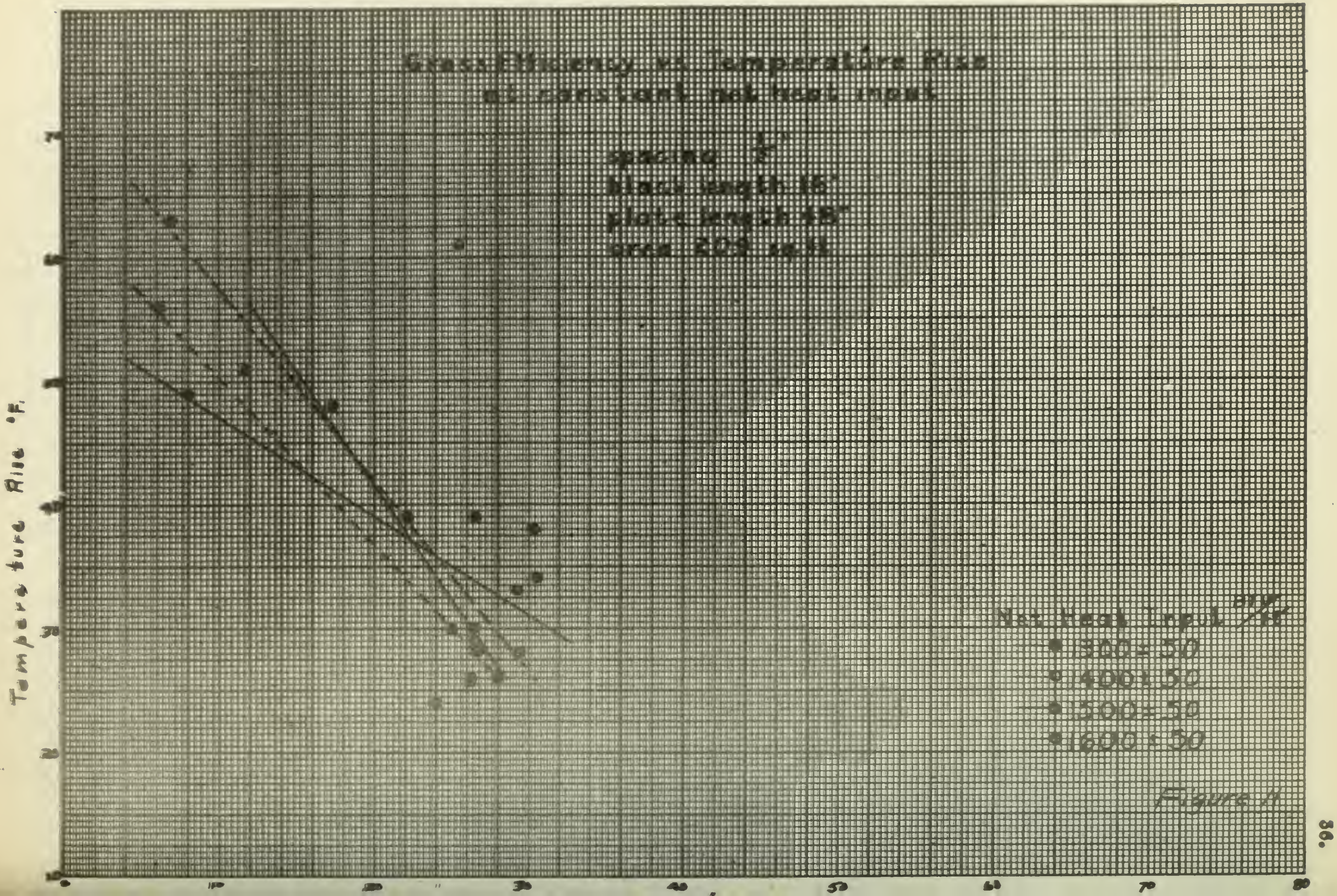

Gross Efficiency vs Temperature Rise
at constant net heat input
spacing 1/4"
black length 16"
plate length 48"
area 2.09 sq.ft.
Net Heat Input, BTU/hr
800 ± 50
1400 ± 50
1700 ± 50
1800 ± 50
Temperature Rise °F.
Gross Efficiency, %
10
20
30
40
50
60
70
80
0
10
20
30
40
50
60
70
80

Figure 10



Gross Efficiency vs. Temperature Rise
at constant net heat input
spacing
blade length 16"
plate length 18"
Temperature Rise °F.
Net Heat Input
300 ± 50
400 ± 50
500 ± 50
600 ± 50
Figure 11

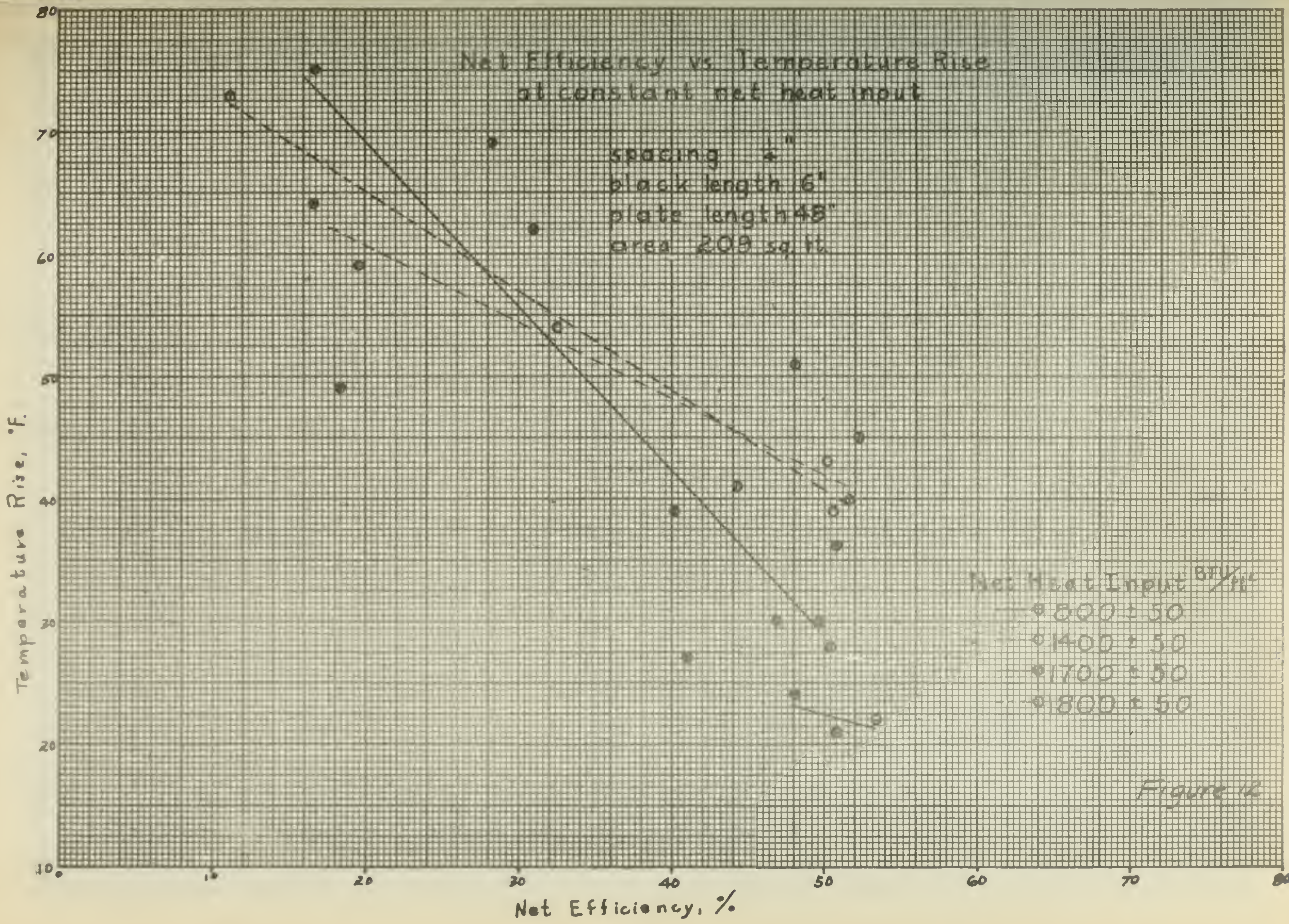
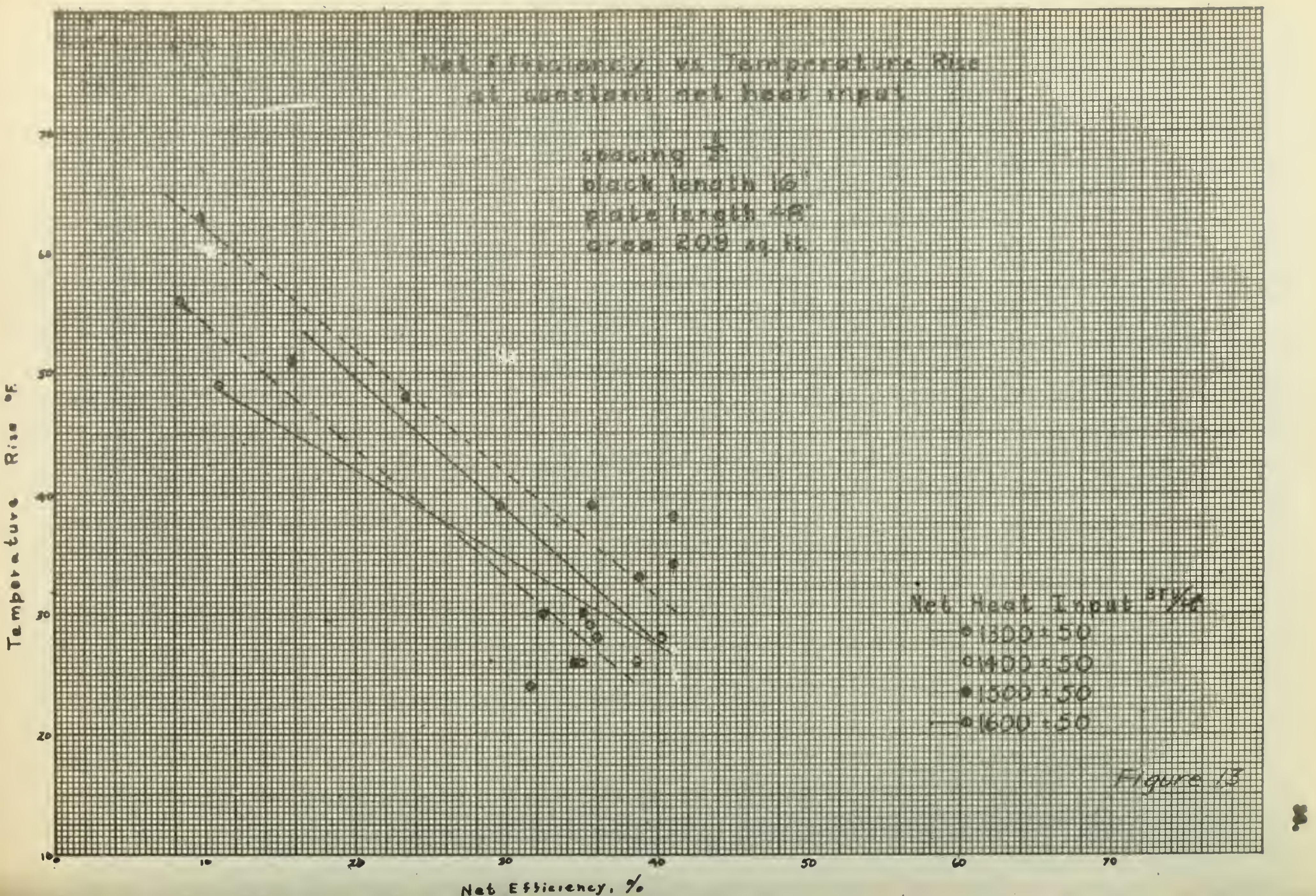

Net Efficiency vs Temperature Rise
at constant net heat input
spacing 1/4"
black length 6"
plate length 48"
area 209 sq. ft.
Net Heat Input BTU/ft²
800 ± 50
1400 ± 50
1700 ± 50
1800 ± 50
Temperature Rise, °F.
Net Efficiency, %
80
70
60
50
40
30
20
10
0
10
20
30
40
50
60
70
80

Figure 12



Net Efficiency vs Temperature Rise
at constant net heat input
spacing $\frac{1}{2}$
black length 16"
plate length 48"
area 209 sq ft
Net Heat Input BTU/hr
1300 ± 50
1400 ± 50
1500 ± 50
1600 ± 50
Figure 13
Temperature Rise °F
Net Efficiency, %
10
20
30
40
50
60
70

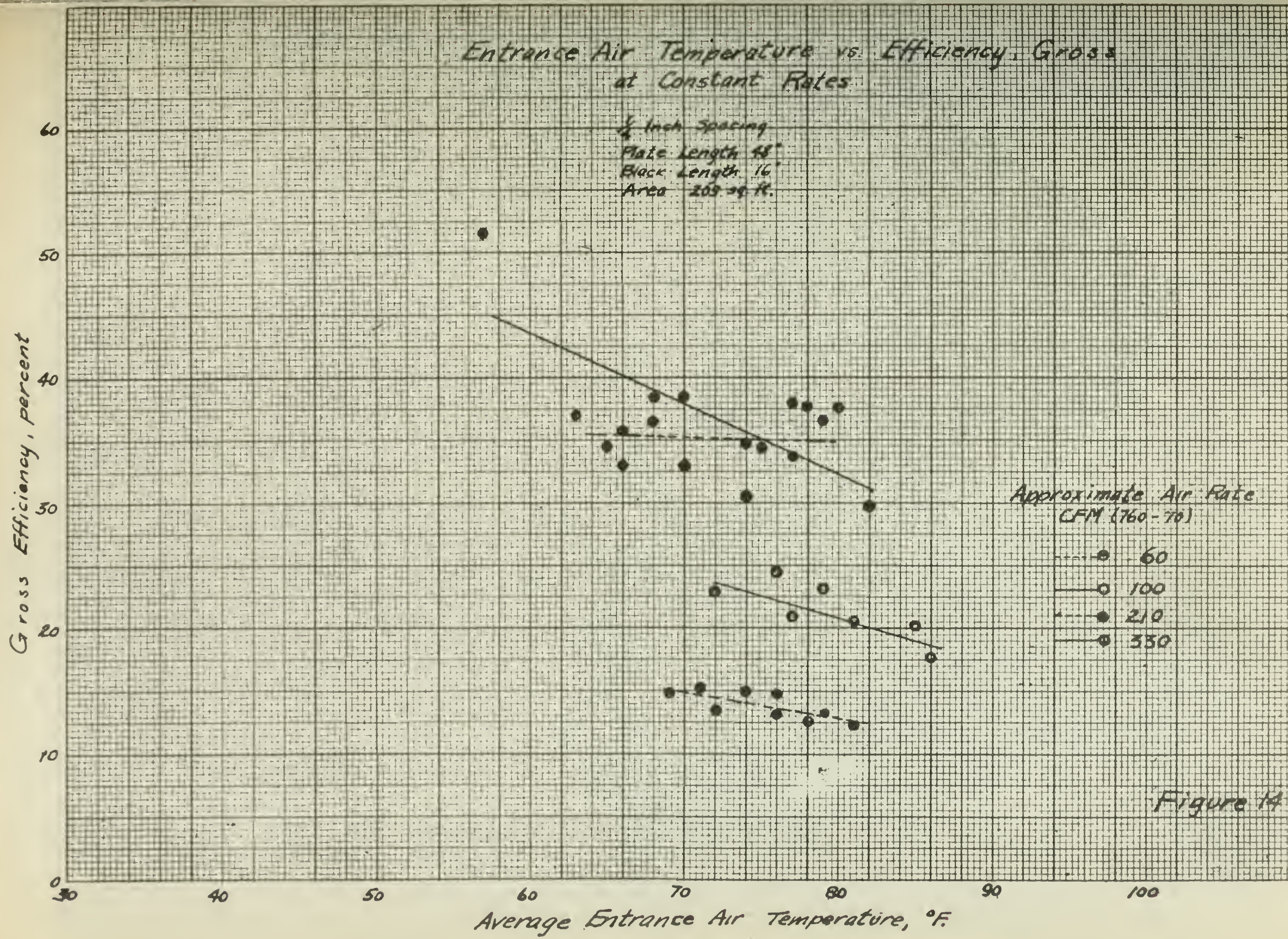

Entrance Air Temperature vs. Efficiency, Gross
at Constant Rates
1/4 Inch Spacing
Plate Length 48"
Black Length 16"
Area 263 sq ft.
Approximate Air Rate
CFM (760 - 70)
60
100
210
330
Gross Efficiency, percent
0
10
20
30
40
50
60
30
40
50
60
70
80
90
100
Average Entrance Air Temperature, °F.

Figure 14

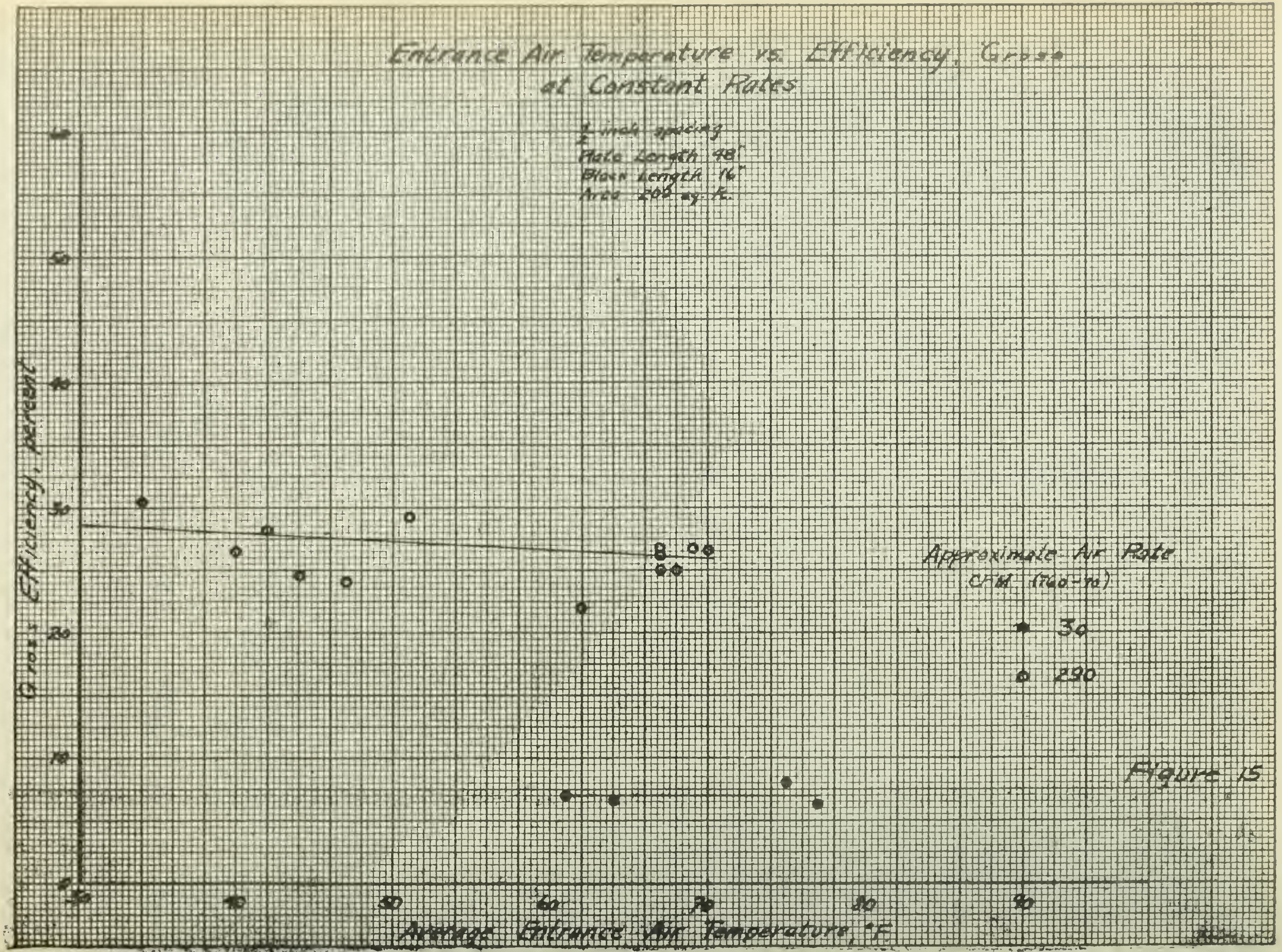

Entrance Air Temperature vs. Efficiency, Gross
at Constant Rates
½ inch spacing
Plate Length 48"
Blower Length 16"
Area 200 sq. ft.
Approximate Air Rate
CFM (760-70)
30
290
Gross Efficiency, percent
Average Entrance Air Temperature, °F
10
20
30
40
50
60
30
40
50
60
70
80
90

Figure 15

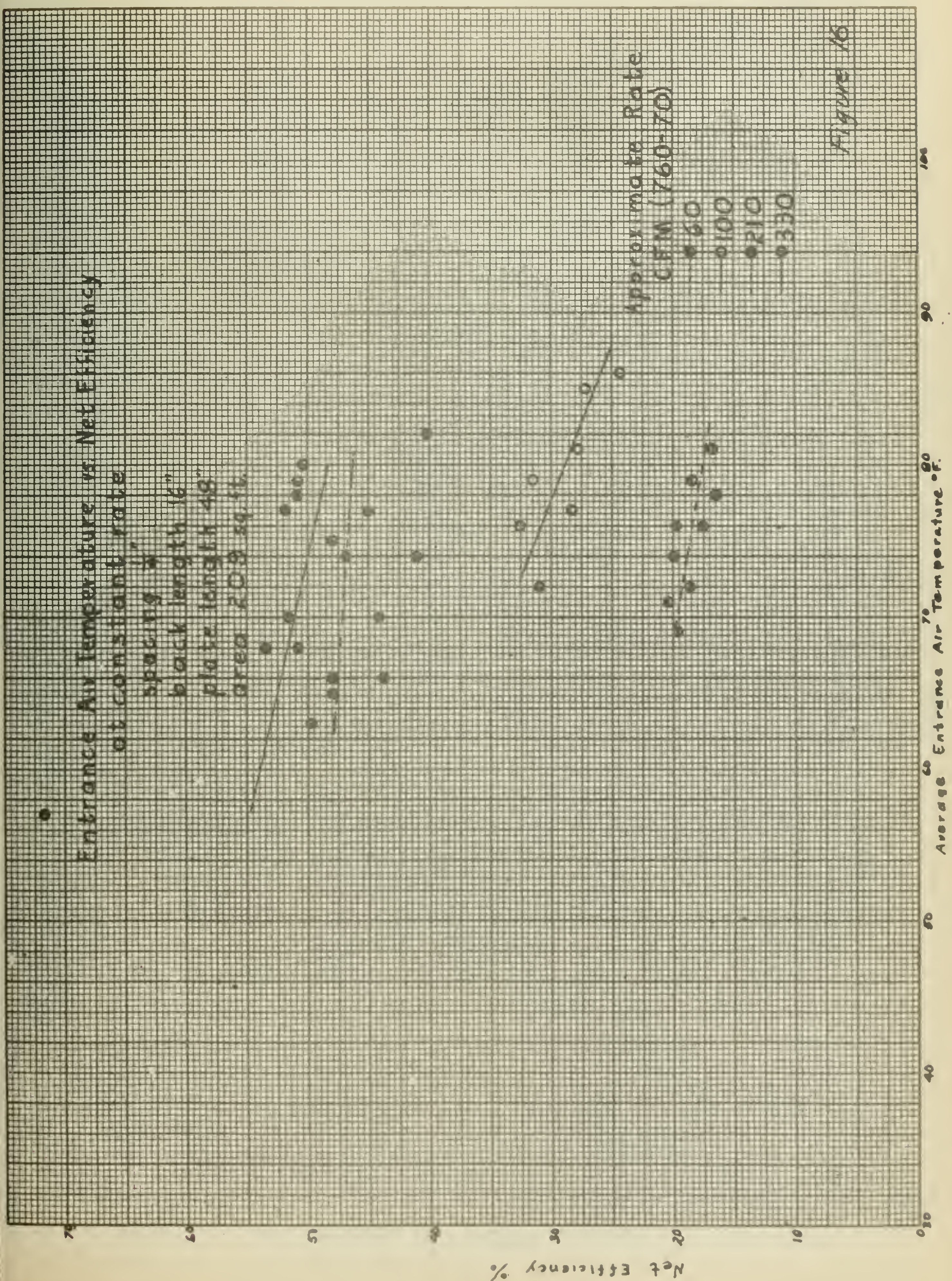
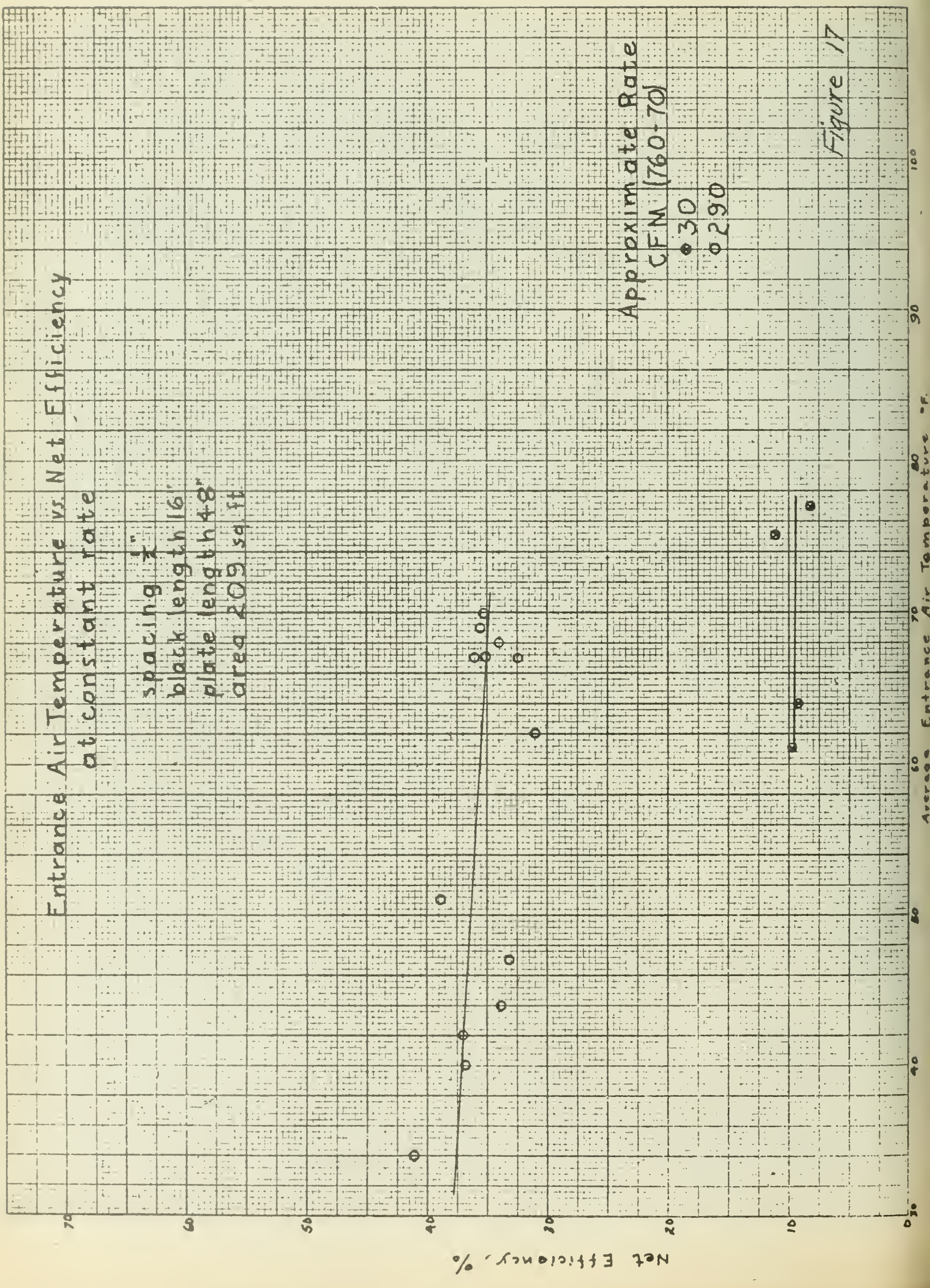
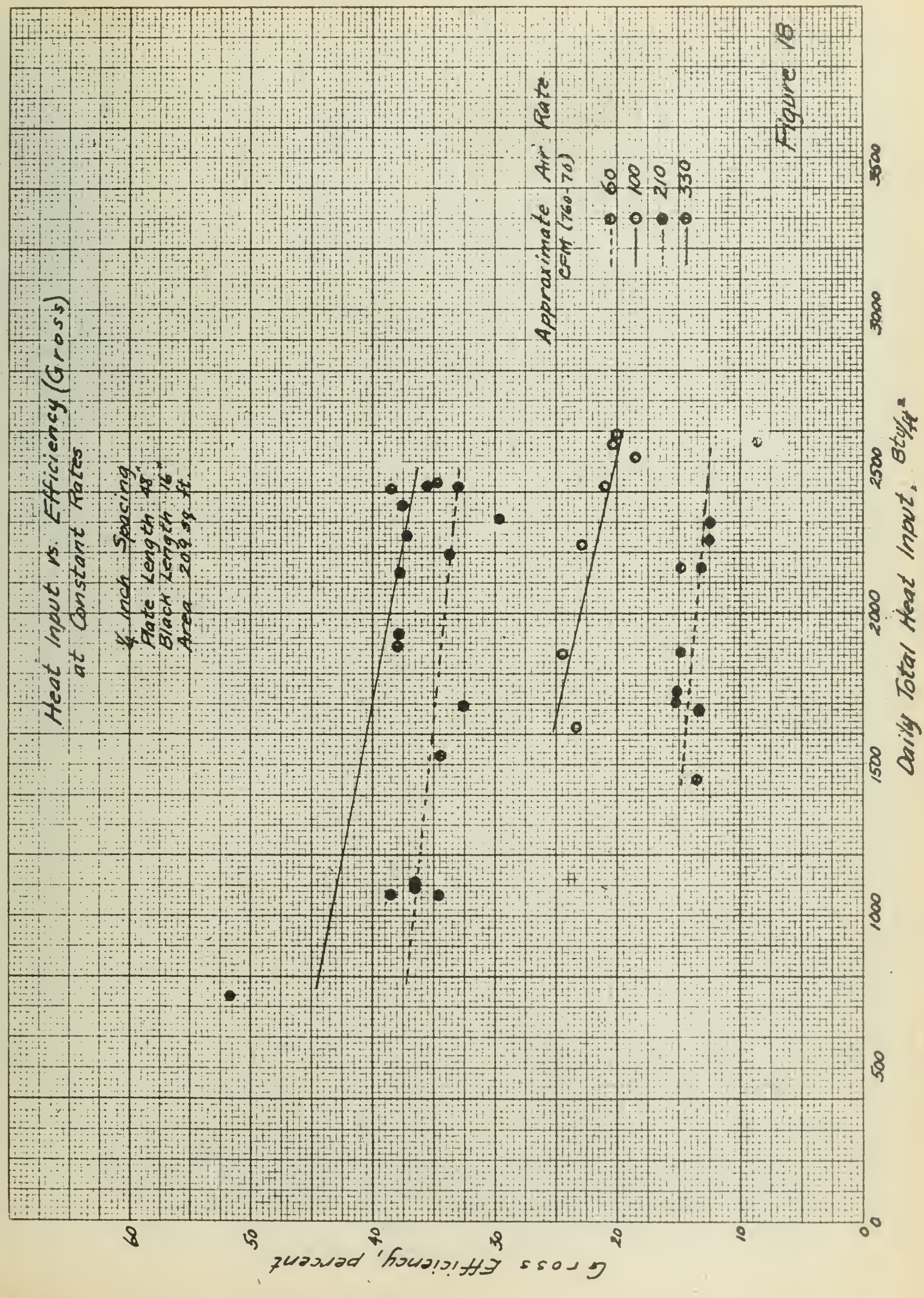
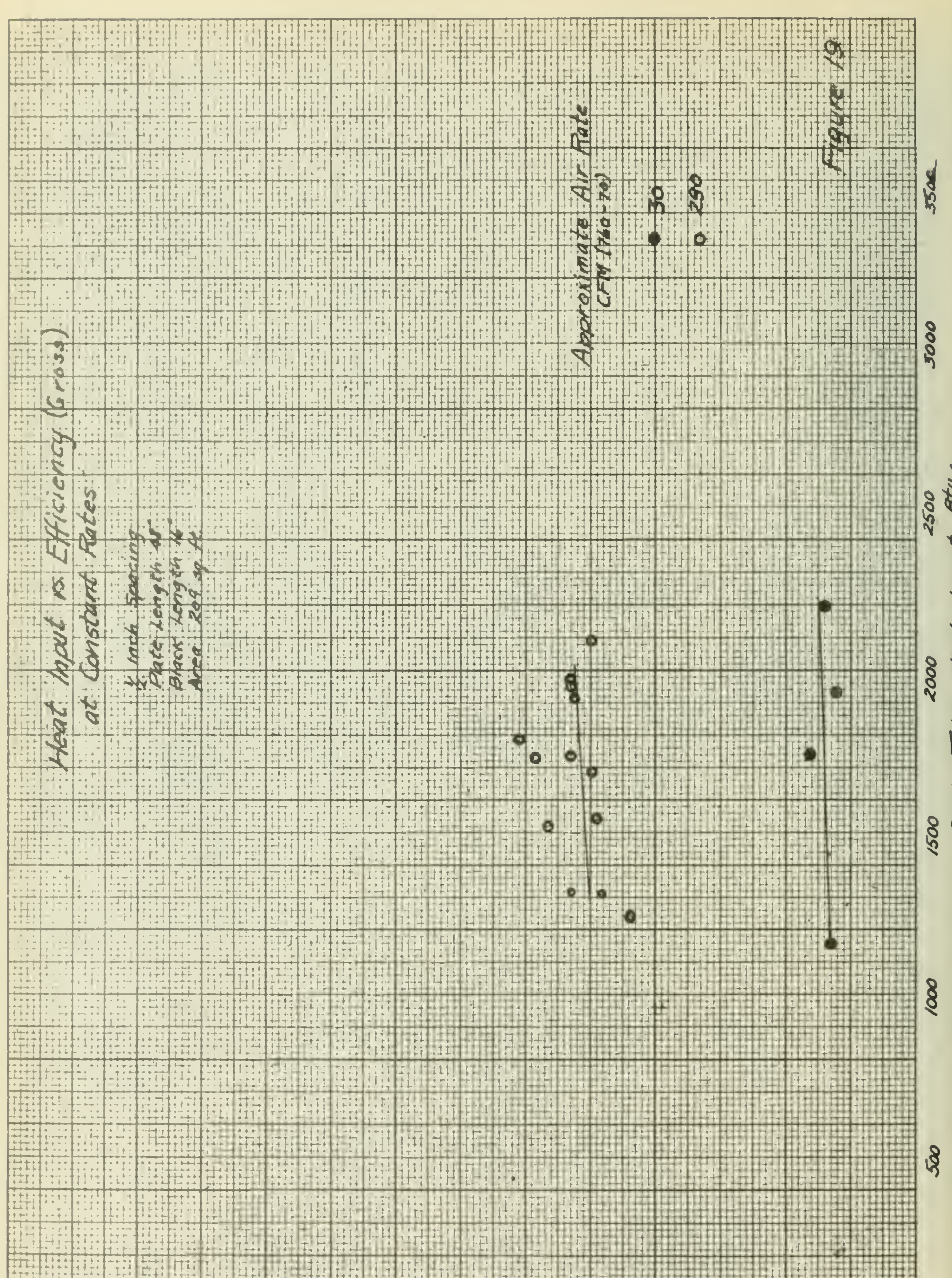

Entrance Air Temperature vs. Net Efficiency
at constant rate
spacing 1"
black length 16"
plate length 48"
area 2.09 sq.ft.
Approximate Rate
CFM (760-70)
60
100
210
330
Average Entrance Air Temperature °F.
Net Efficiency %
30
40
50
60
70
80
90
100
0
10
20
30
40
50
60
70

Figure 16


Entrance Air Temperature vs. Net Efficiency
at constant rate
spacing ¾"
black length 16"
plate length 48"
area 2.09 sq. ft.
Approximate Rate
CFM (760-70)
30
290
Figure 17
Average Entrance Air Temperature °F.
Net Efficiency, %
30
40
50
60
70
80
90
100
0
10
20
30
40
50
60
70


Heat Input vs. Efficiency (Gross)
at Constant Rates
1/4 Inch Spacing
Plate Length 48"
Black Length 16"
Area 20.9 sq. ft.
Approximate Air Rate
CFM (760-70)
60
100
210
330
Figure 18
Gross Efficiency, percent
Daily Total Heat Input, Btu/ft²
0
10
20
30
40
50
60
0
500
1000
1500
2000
2500
3000
3500



Heat Input vs Efficiency (Gross)
at Constant Rates
½ inch Spacing
Plate Length
Black Length
Approximate Air Rate
CFM (760-70)
30
290
Figure 19
Gross Efficiency, percent
0
10
20
30
40
50
60
Daily Total Heat Input, Btu/ft²
0
500
1000
1500
2000
2500
3000
3500

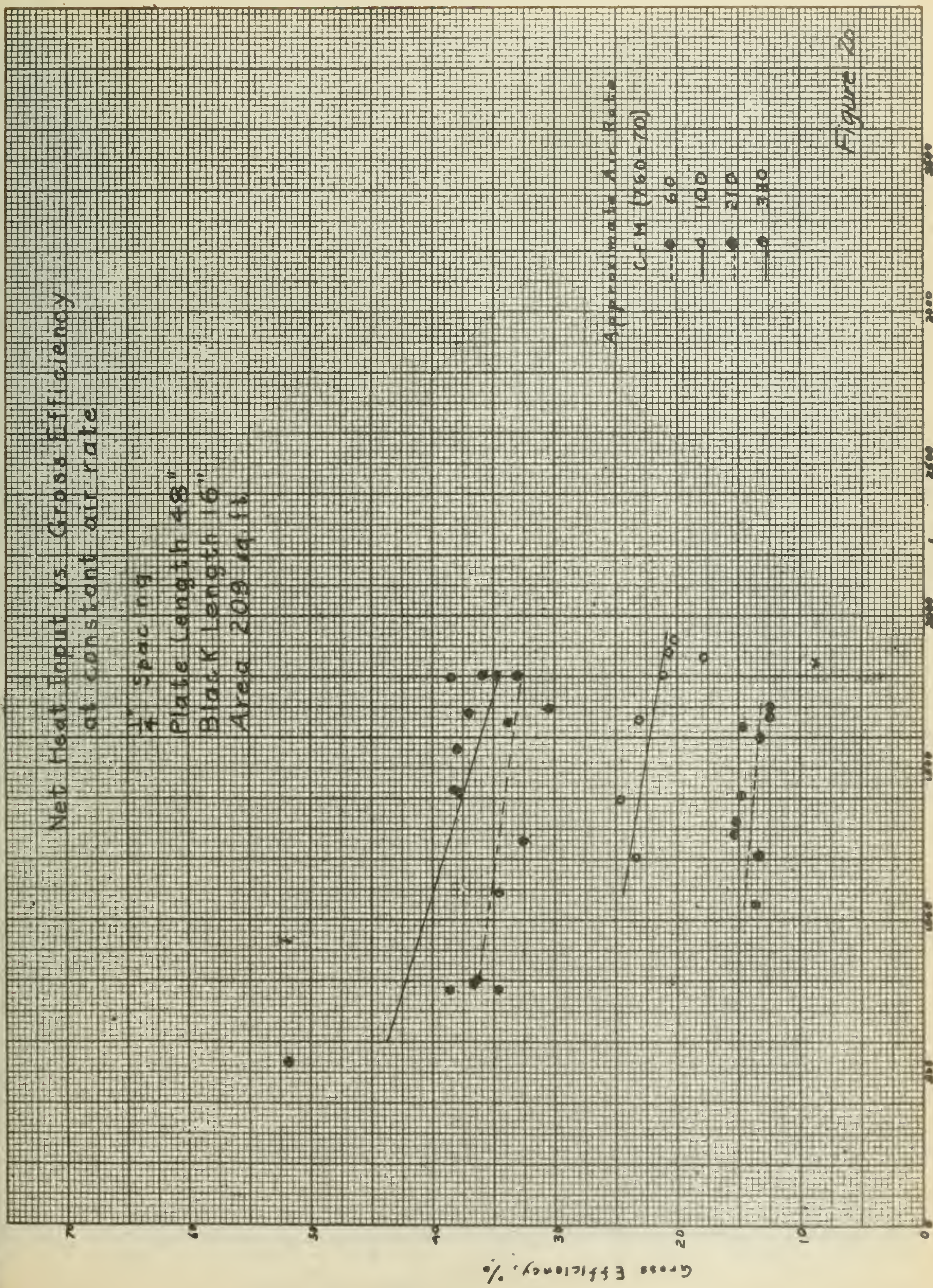
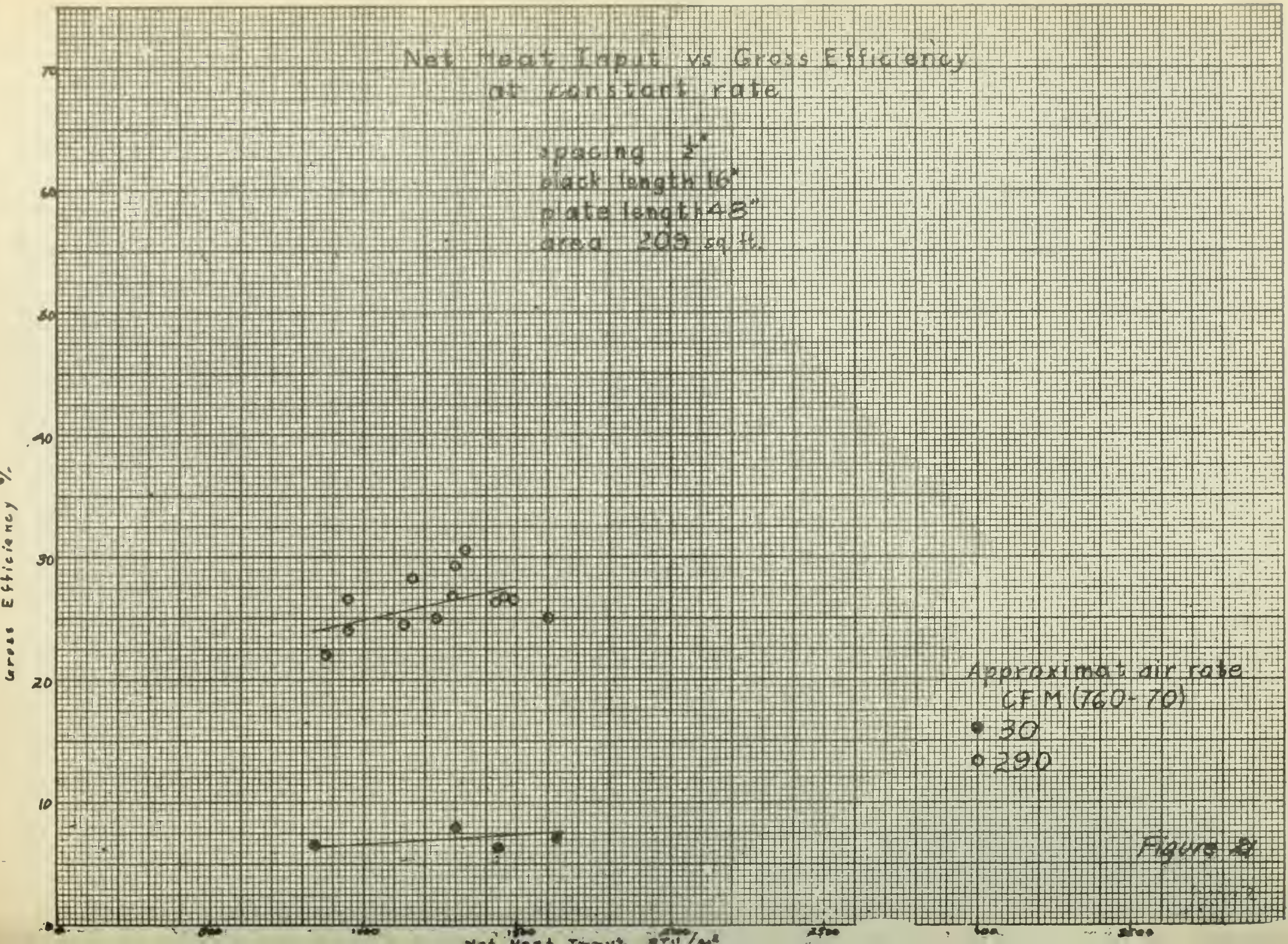

Net Heat Input vs. Gross Efficiency
at constant air rate
1/4" Spacing
Plate Length 48"
Black Length 16"
Area 209 sq ft
Approximate Air Rate
CFM (760-70)
60
100
210
380
Net Heat Input, B.T.U./sq. ft.
Gross Efficiency, %

Figure 20


Net Heat Input vs Gross Efficiency
at constant rate
spacing 1/2"
stack length 16"
plate length 48"
area 209 sq ft.
Approximat air rate
CFM (760-70)
30
290
Gross Efficiency %
70
60
50
40
30
20
10
Figure

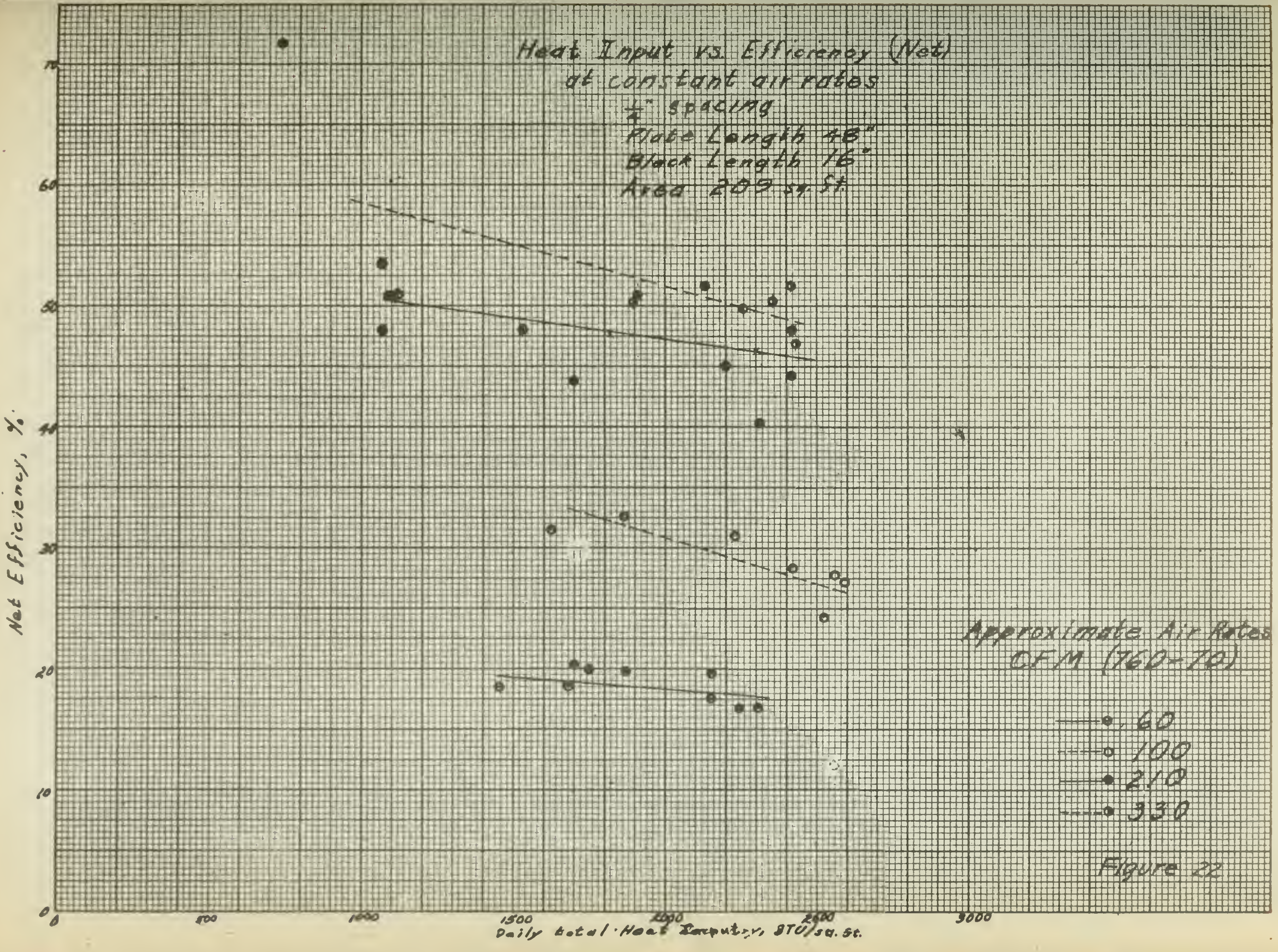
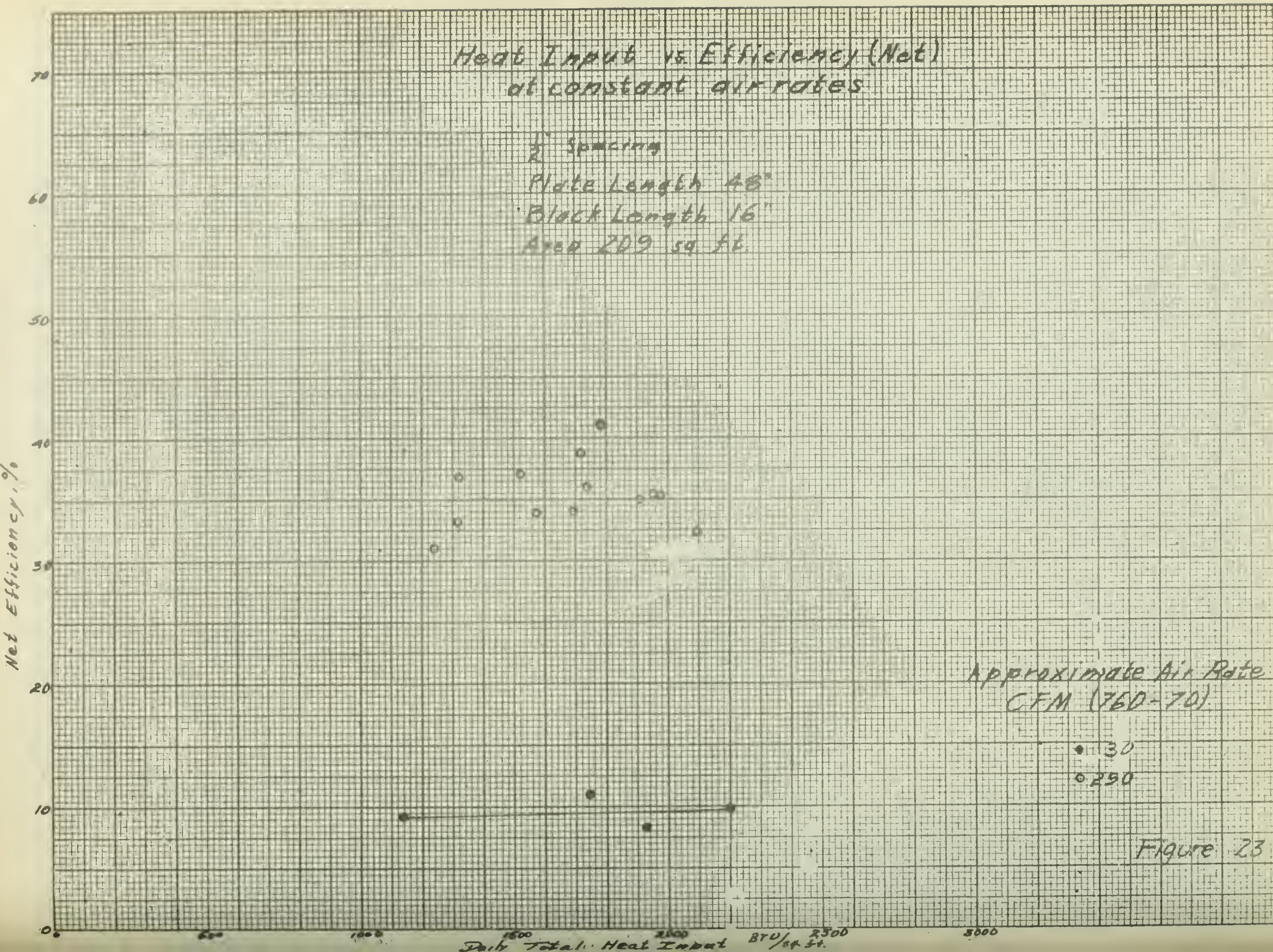

Heat Input vs. Efficiency (Net)
at constant air rates
1/4" spacing
Plate Length 48"
Black Length 16"
Area 209 sq. ft.
Net Efficiency, %.
Daily total Heat Input, BTU/sq. ft.
Approximate Air Rates
CFM (760-70)
60
100
210
330
0
10
20
30
40
50
60
70
500
1000
1500
2000
2500
3000

Figure 22


Heat Input vs Efficiency (Net)
at constant air rates
1/2" Spacing
Plate Length 48"
Black Length 16"
Area 209 sq ft
Approximate Air Rate
CFM (760-70)
30
250
Figure 23
Net Efficiency, %
70
60
50
40
30
20
10
0
500
1000
1500
2000
2500
3000
Daily Total Heat Input BTU/sq ft.

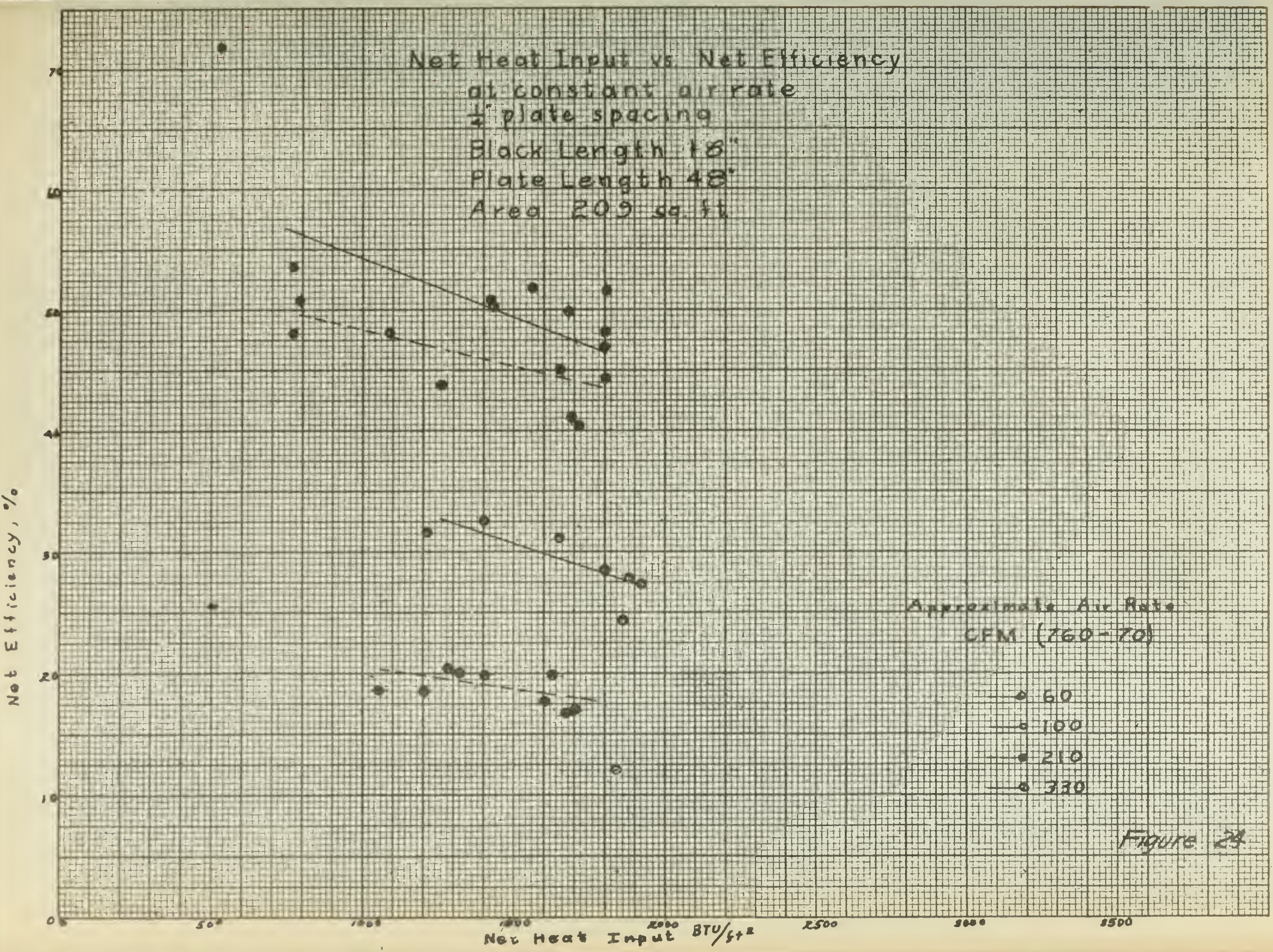
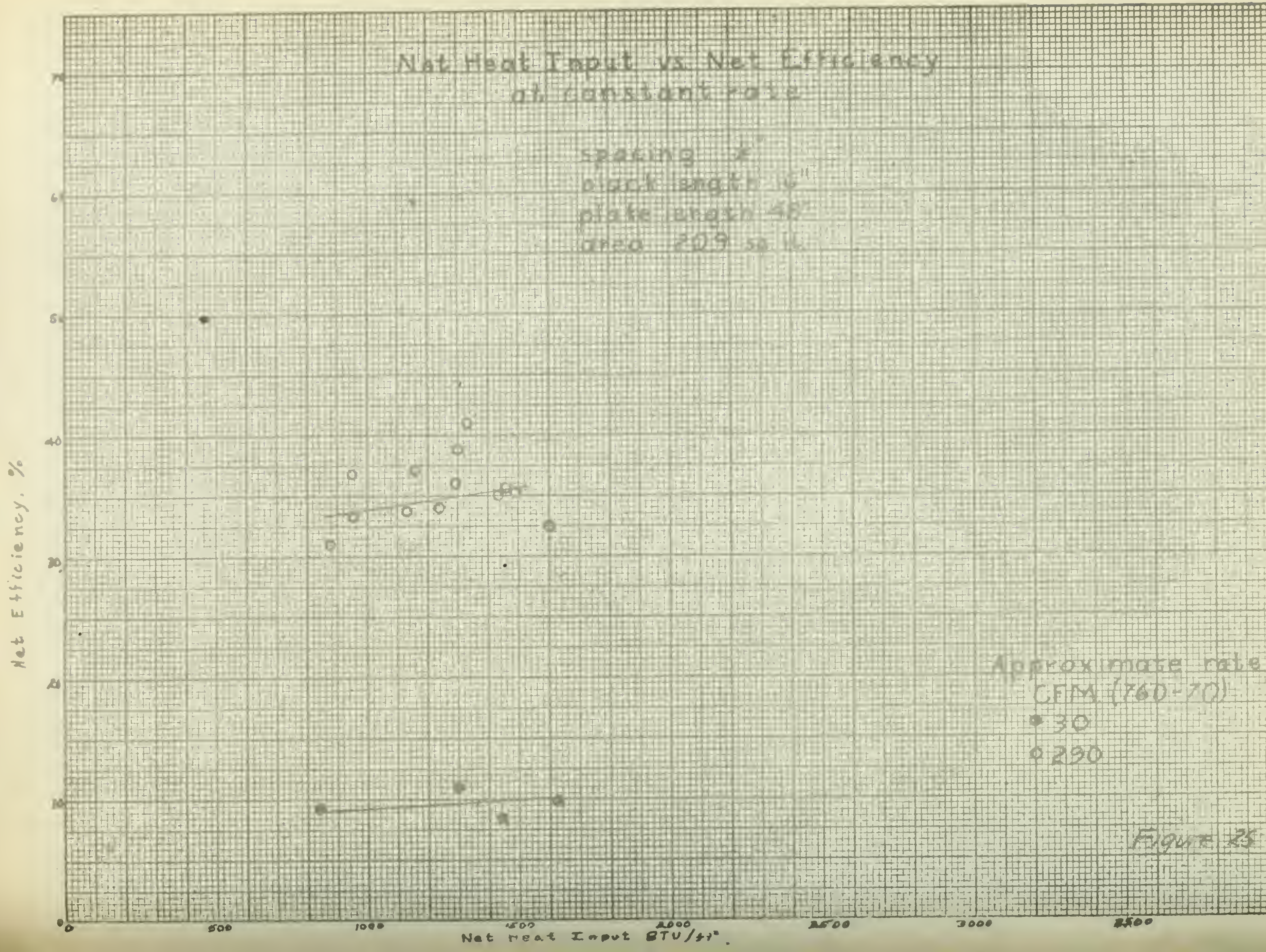

Net Heat Input vs Net Efficiency
at constant air rate
1/4" plate spacing
Black Length 18"
Plate Length 48"
Area 209 sq. ft.
Approximate Air Rate
CFM (760-70)
60
100
210
330
Net Efficiency, %
Net Heat Input BTU/ft²
0
10
20
30
40
50
60
70
500
1000
1500
2000
2500
3000
3500

Figure 24


Net Heat Input vs Net Efficiency
at constant rate
spacing
block length
plate length 48"
area 209 sq in
Approximate rate
CFM (760-70)
30
290
Figure 25
Net Efficiency, %
Net Heat Input BTU/ft²
0
10
20
30
40
50
60
70
500
1000
1500
2000
2500
3000
3500

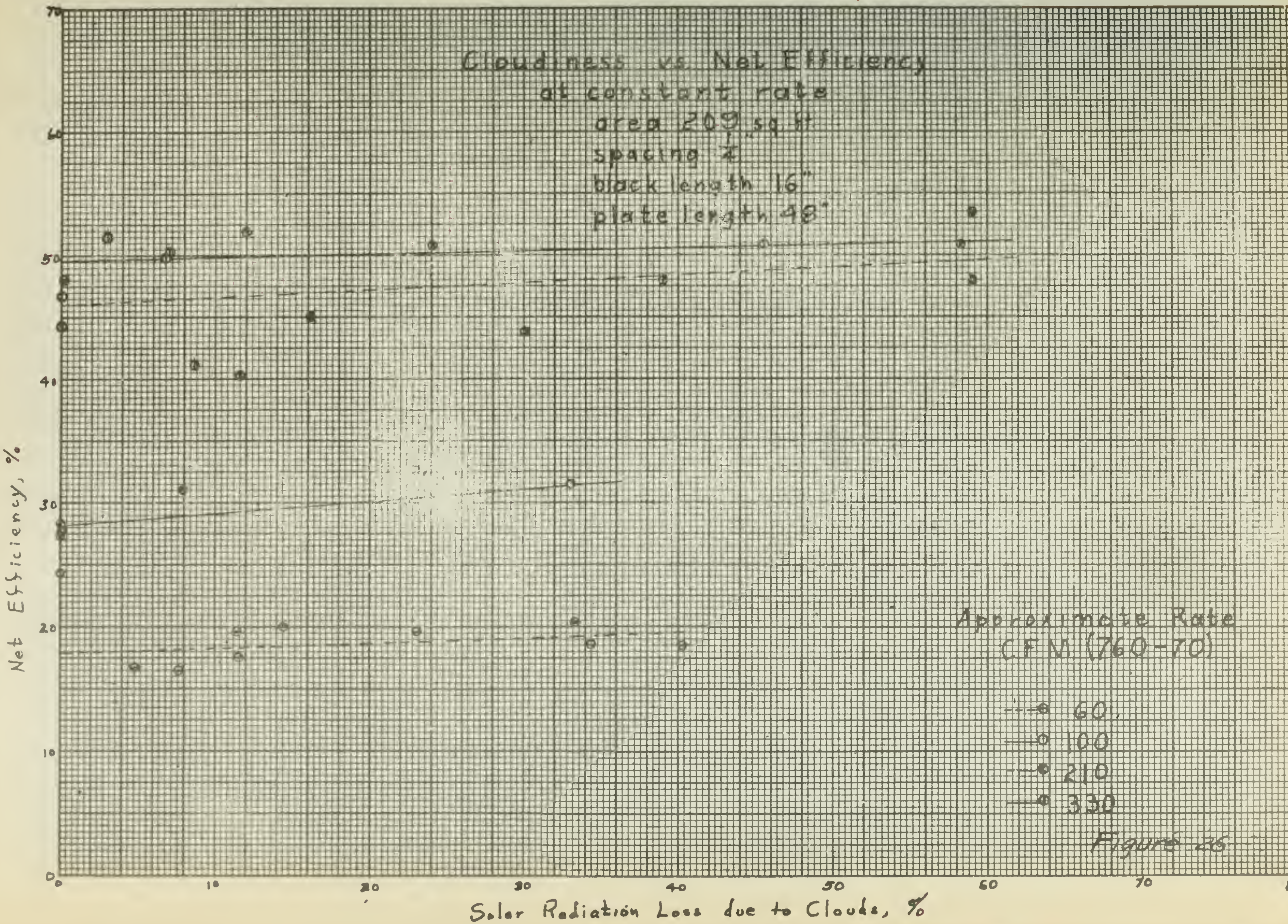
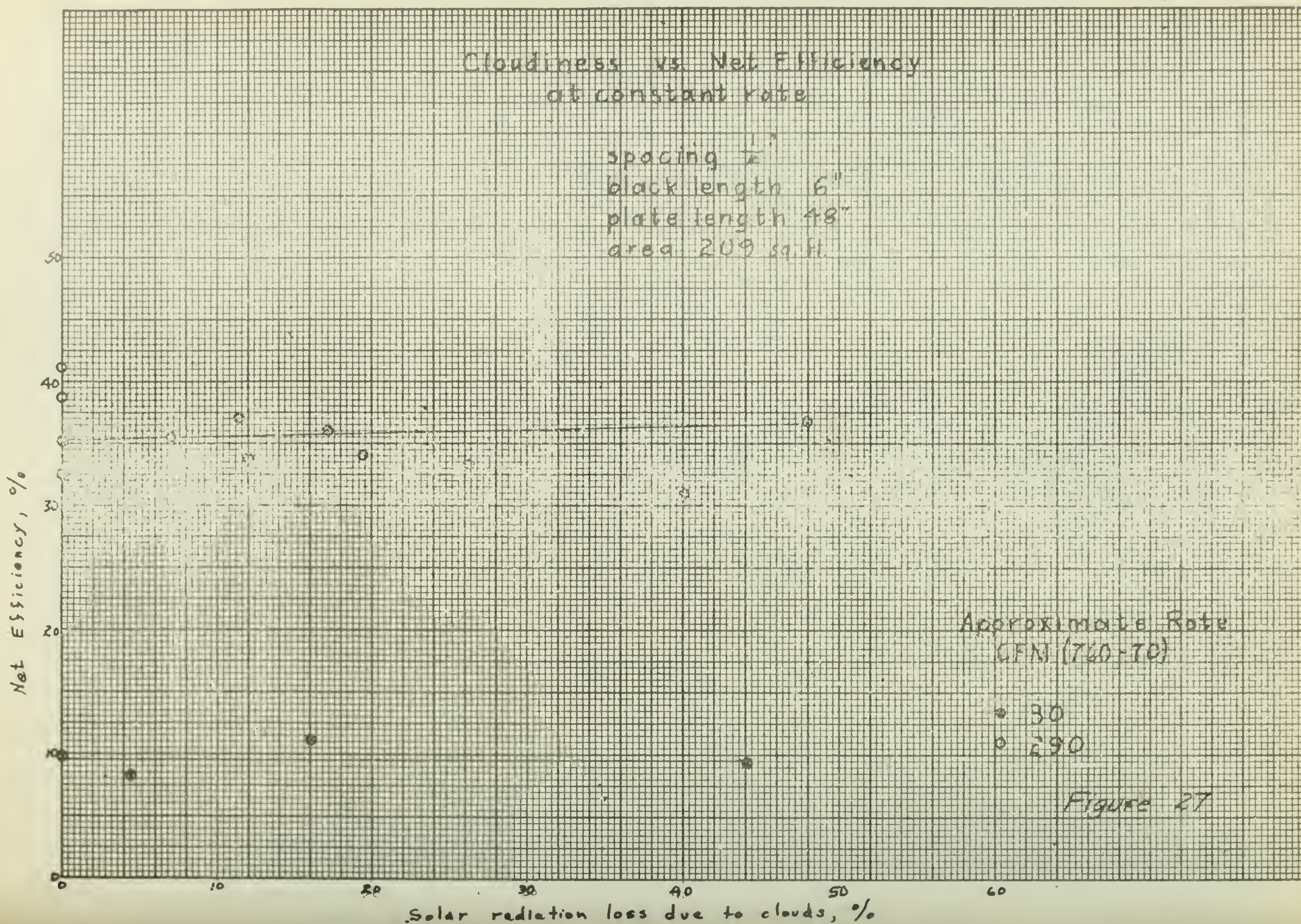
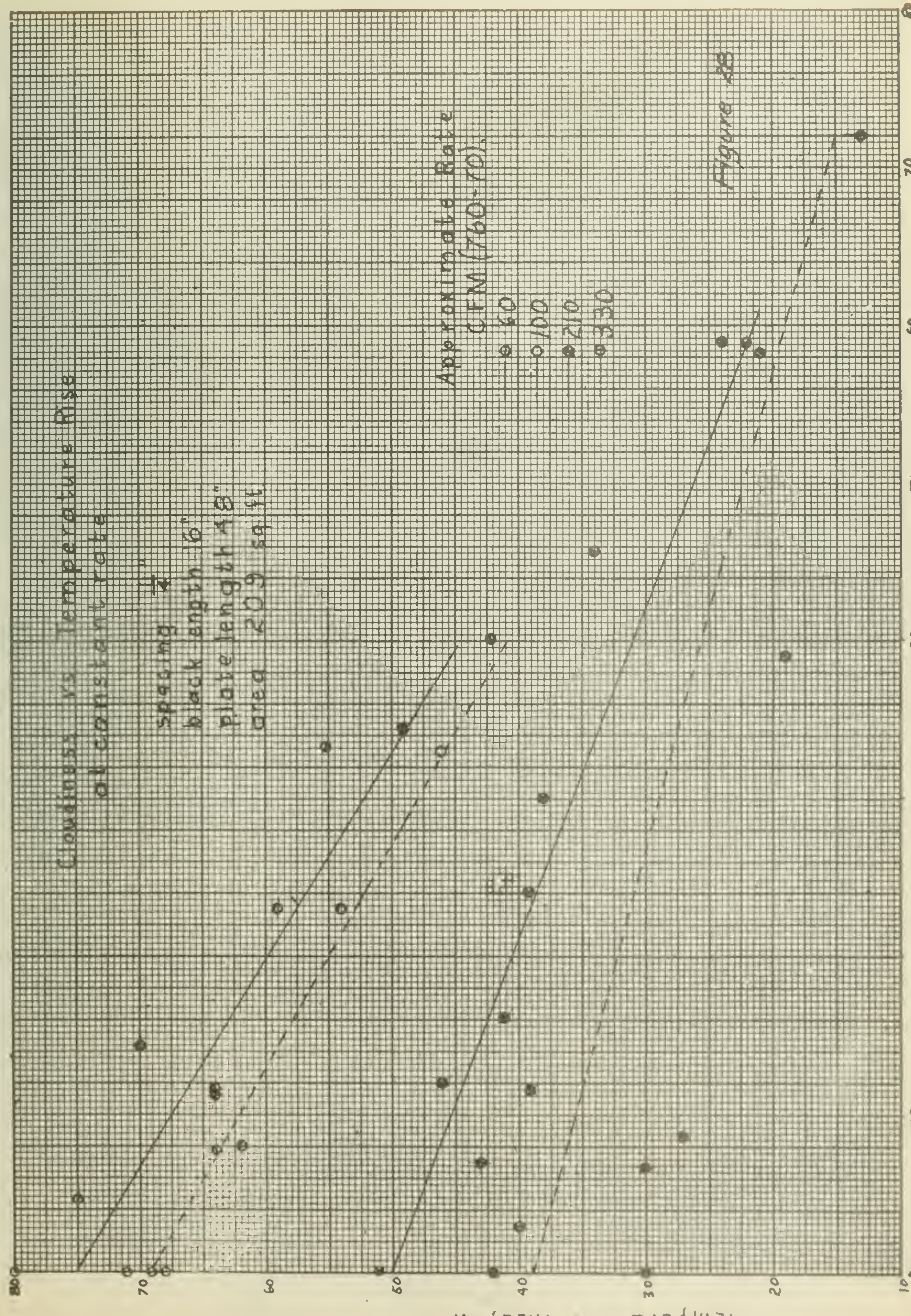
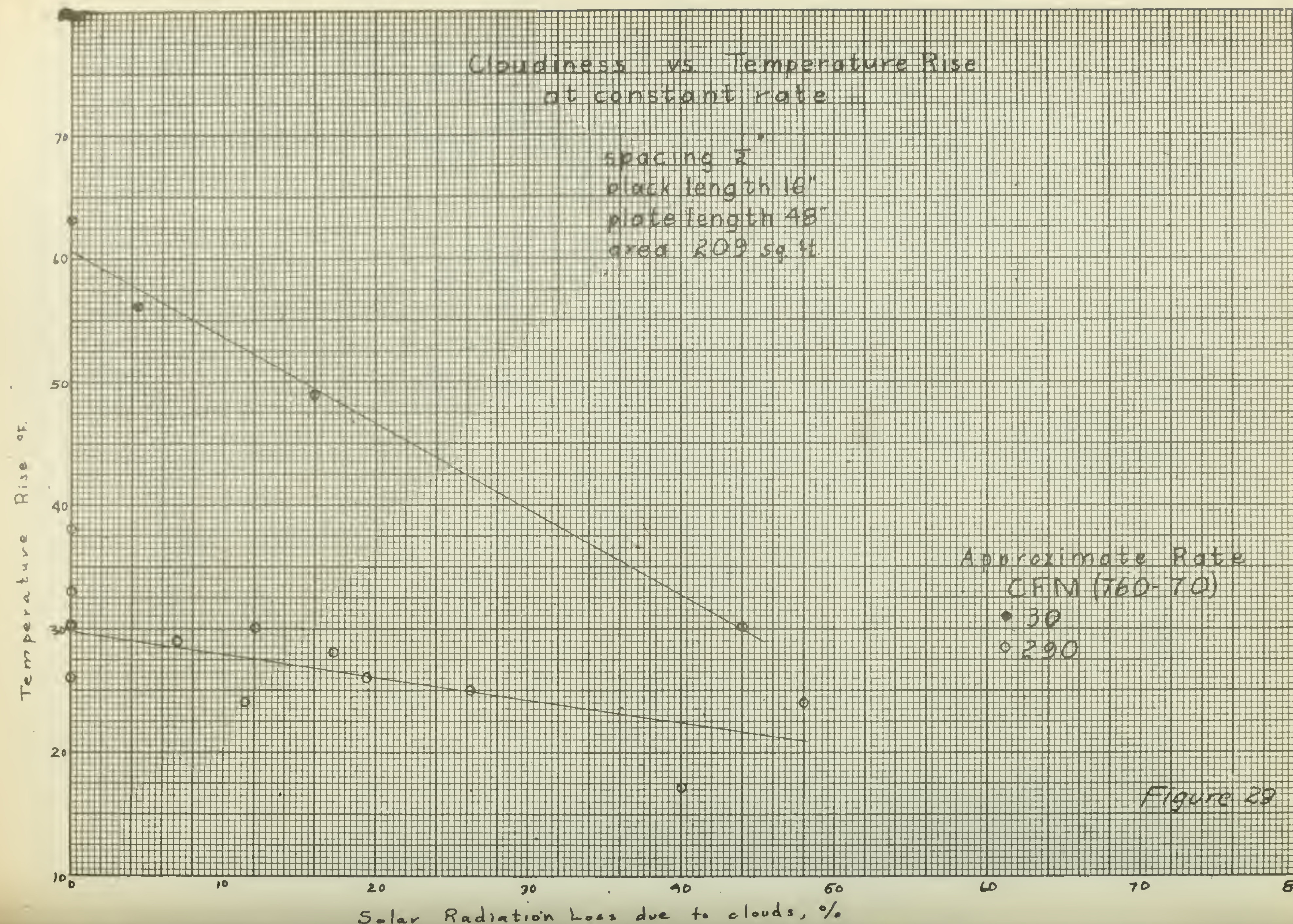

Cloudiness vs Net Efficiency
at constant rate
area 209 sq ft
spacing ¼"
black length 16"
plate length 48"
Approximate Rate
CFM (760-70)
60
100
210
330
Net Efficiency, %
Solar Radiation Loss due to Clouds, %
0
10
20
30
40
50
60
70
80

Figure 26



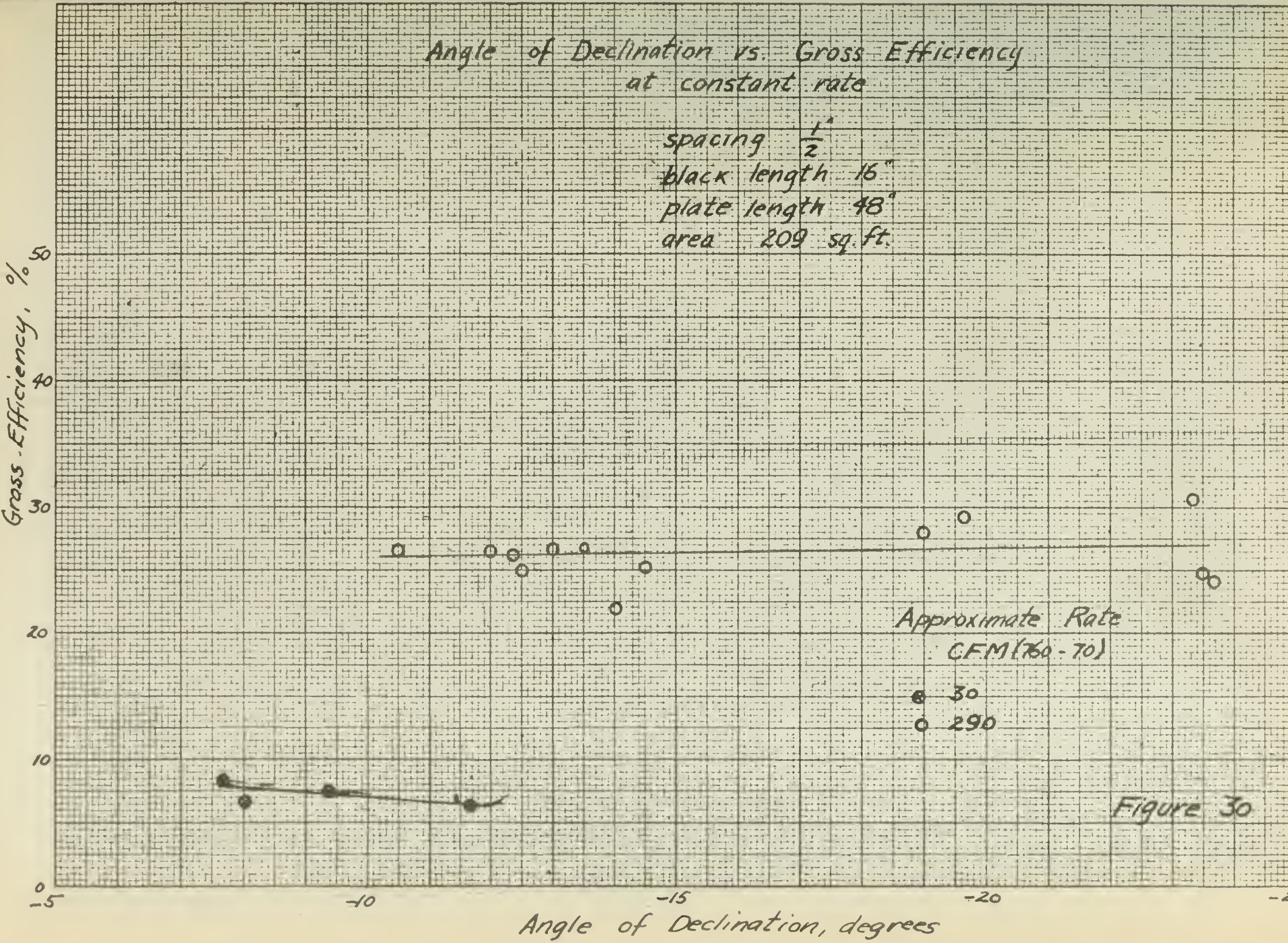
Cloudiness vs. Net Efficiency
at constant rate
spacing 1/4"
black length 6"
plate length 48"
area 209 sq ft
Approximate Rate
CFM (740-70)
30
290
Figure 27
Net Efficiency, %
Solar radiation loss due to clouds, %
0
10
20
30
40
50
60


Cloudiness vs. Temperature Rise
at constant rate
spacing 1/4"
black length 6"
plate length 48"
area 209 sq ft
Approximate Rate
CFM (760-70).
60
100
210
330
Figure 28
Temperature Rise, °F
Solar Radiation Loss due to clouds %
10
20
30
40
50
60
70
80
0
10
20
30
40
50
60
70
80



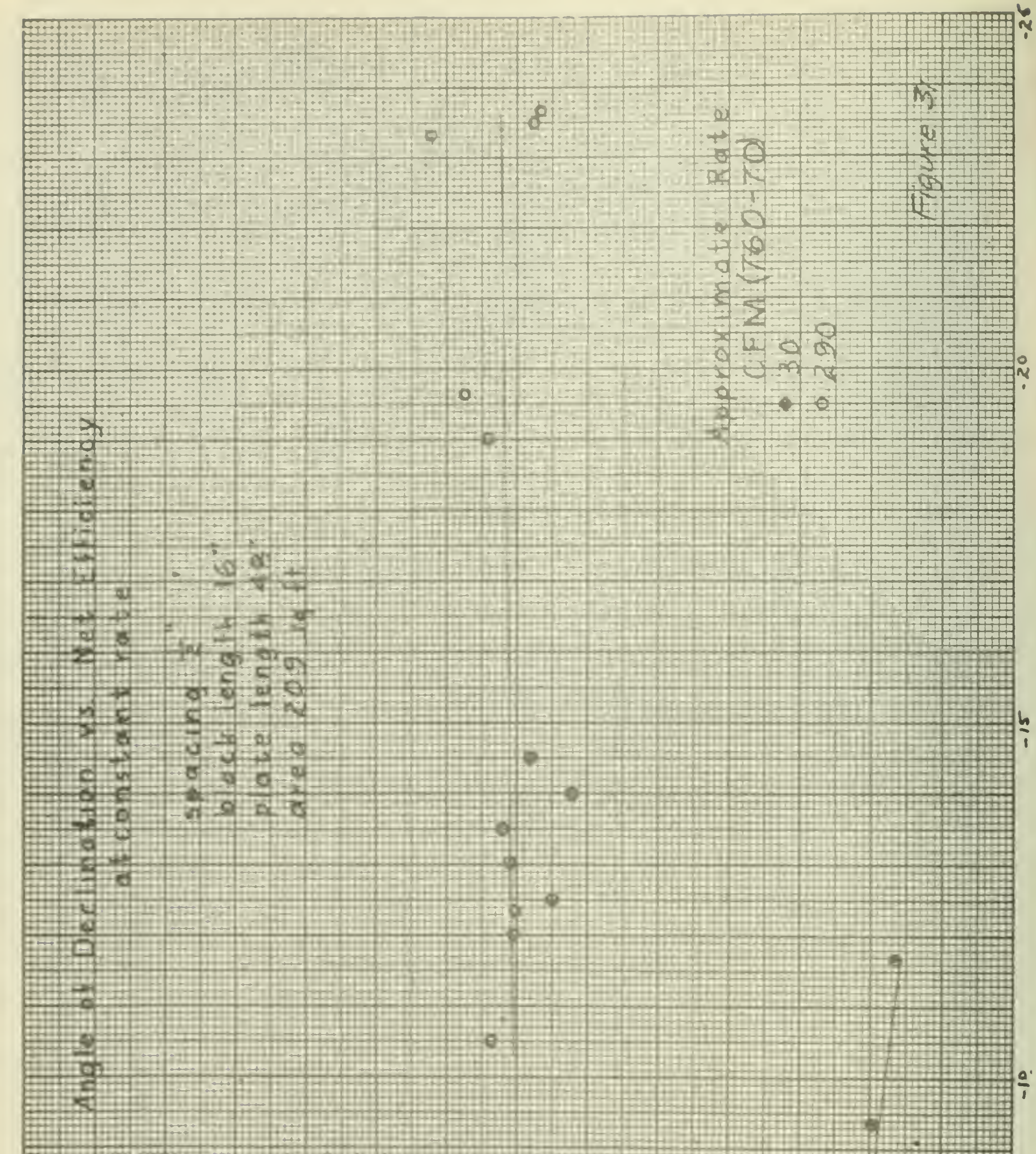
Cloudiness vs. Temperature Rise
at constant rate
spacing ½"
black length 16"
plate length 48"
area 209 sq ft
Approximate Rate
CFM (760-70)
• 30
○ 290
Temperature Rise °F.
Solar Radiation Loss due to clouds, %
10
20
30
40
50
60
70
0
10
20
30
40
50
60
70
80

Figure 29



Angle of Declination vs. Gross Efficiency
at constant rate
spacing 1/2"
black length 16"
plate length 48"
area 209 sq. ft.
Approximate Rate
CFM (760 - 70)
30
290
Gross Efficiency, %
0
10
20
30
40
50
-5
-10
-15
-20
-25
Angle of Declination, degrees

Figure 30


Angle of Declination vs. Net Efficiency
at constant rate
spacing 1/2"
black length 16"
plate length 42"
area 209 sq ft
Approximate Rate
CFM (760-70)
• 30
○ 290
Figure 31
Net Efficiency %
0
10
20
30
40
50
-5
-10
-15
-20
-25

VII GENERAL SIGNIFICANCE OF RESULTS

One of the most important results of the entire investigation is the proof f the workability of the principle advanced by Miller (10). Although experimental eat recovery efficiencies and exit air temperatures are not as high as predicted heoretically, the results indicate that solar energy can be used to heat air to emperatures in excess of 200°F and that overall heat recovery efficiencies of at east 35% of total input can be secured.

Of perhaps the greatest practical significance is the fact that a solar heatng unit was constructed and installed in a typical small home; was provided with ompletely automatic controls; was tied in with the normal heating system; and as operated satisfactorily throughout one complete winter. A fuel saving of pproximately 20 per cent was realized.

Other important specific results involve the details of construction and operaion of the equipment. Such factors as the use of two-thirds overlap of plates ainted one-third black on the upper surface, $\frac{1}{4}$ inch spacing between plates, plate engths of three to four feet, support of plates on strips driven into wood section ides, and method of supporting cover plates, are practical results of building nd testing the equipment. The recirculation of house air through the unit, aintenance of air rate at the optimum value, available temperature of air, heatng load carryable, equalization of temperature in the various sections, are useul results obtained from operating studies on the equipment.

The results indicate that there are good future possibilities for utilization f this solar heating principle, and that further work to develop the apparatus nto commercially marketable units is justified. Indications are that with an expenditure in the general range of $500, the owner of a moderately sized house could ave solar equipment installed which would save two-thirds of his winter fuel in climate having considerable winter sunshine, but rather large heating requireents. The relatively low cost and simplicity of construction of the equipment re probably its greatest advantages; and since the efficiencies are comparable o those secured with flat plate collectors heating water (9), and much greater han those obtained with simple large south windows, it is felt that commercial evelopment may be possible as soon as the few remaining problems have been solved.

It is to be realized that all data, calculations, and results have been based n observations made at one location, Boulder, Colorado, 40 degrees 0 min North atitude, 105 degrees 16 min West longitude, 5428 feet elevation. All the results n general, and those of the house operation in particular cannot be applied to ther locations without discretion. Certain generalizations can be made however. hus, in locations in which either the solar radiation received is greater than n Boulder (because of lower latitude, for example) or the heating requirement s less (because of higher winter temperatures), either the fraction of the heating oad carryable by a solar unit of the same size would be greater in the other locaion, or a collector to carry the same load could be smaller or tilted at a lower ngle. For prediction of size of collector needed to carry a certain heating load n another location, data on latitude, solar radiation, winter temperatures, and ouse heating requirements are necessary.

Results obtained with the collector alone, as in the case of the laboratory runs, can be applied more readily to other locations; and comparable performance can be expected if consideration is given to the different angle at which the sun strikes the collector at the different latitude. Thus, for the same projected area of collector, and equivalent average cloudiness, heat recovery efficiencies should be very similar at the same air rates, even in widely different locations.

It is probable that although the conditions for utilizing solar energy at Boulder, Colorado are considered to be good, they are close to the severity beyond which it would be uneconomical to attempt solar heating. It is therefore felt that in localities north of the fortieth parallel, the cost of heating house by solar radiation would generally be too high for economical application of this principle. Collector and storage units would have to be unduly large and their cost would be a considerable drawback. It is possible, however, that in a few northern localities, where considerable winter sunshine is obtained, and where temperatures are not extremely low, economical use might extend considerably above the fortieth parallel. This would be particularly true where fuel is expensive. As further explained below, the greatest applicability of this solar energy collector should be in the southern portion of the United States.

Certain considerations of other uses of the collected solar energy have also been made. The first of these involves the possibility of using the heated air to operate an absorption type of refrigerating air conditioner. Such a unit is available commercially, and the application of solar energy to its operation is considered to have excellent possibilities. It is apparent that the period in which the greatest energy is received corresponds exactly to the period in which the greatest cooling is required. The apparatus would therefore be able to provide the most cooling at the time it is needed.

It is felt that perhaps the region of greatest applicability of this equipmen is in the southern half of the U.S., where winter heating loads are not too sever and where summer cooling is greatly desired.

The heating of water for household uses has also been considered, and equipment for this purpose has been installed in the house unit. No complete results of its operation are available because of its late installation, but a few observations show that, particularly in the summer, it should be possible by using solar-heated air to heat sufficient water to the temperature required for household use.

Other possible uses for solar energy might be to provide a continuous supply of clean, heated air or other gas to certain industrial processes, such as drying or crystallizing.

Numerous factors have not as yet been investigated because of certain limitations. The most important of these problems is that of heat storage. Most of th envisioned uses for the solar energy recoverable by this process are predicated on the assumption that storage of the heat for at least 24 hours can be successfully and economically provided. This problem has not yet been investigated experimentally, because it was considered beyond the scope of the study covered by this report. It is expected , however, that studies now being planned will sh that storage of heat can be accomplished satisfactorily. On this assumption, resu of heat storage calculations have been included in this report. They indicate th

storage of heat only overnight in a bed of crushed solid, will more than double the heating load carryable by the solar unit, and will not require a storage unit too large for practical use. An additional benefit, not yet determined quantitatively, is the higher storage temperatures which can undoubtedly be obtained when hot air is recirculated between collector and storage unit.

An important practical problem not yet solved, because of lack of time, is the discovery of the fundamental cause of, and the method of eliminating, the excessive breakage of glass in the collecting unit. The frequent breakage of plates exposed to sunlight is no doubt caused by thermal stresses set up in the glass, but the way in which these stresses cause breakage is not definitely known. Tests are being planned which should show whether small edge cracks grow into large ones, whether lack of proper annealing is responsible, or if some other cause exists.

Although considerable progress has been made toward designing a collector entirely practical for the average home, more work needs to be done to make possible the large scale production of units which then can be easily installed in the house. Certain simplification and standardization in design is necessary before wide usage can be realized.

Another subject meriting study is the possible use of glass which has been surface-treated to reduce reflection. It would perhaps be possible, at comparatively low cost, to reduce materially the amount of energy reflected back to the sky from the unit, and thereby increase the efficiency.

The black coating used on plates in all the tests was very satisfactory. It had a high absorbtivity for solar radiation and reasonably good adherence. Future work, however, will be directed toward the development of a coating with extremely permanent characteristics and one which could be easily applied at the glass factory.

Weather resistance of the equipment was generally very good. The use of putty for sealing the cover glasses on the house unit proved very satisfactory, except when it was necessary to replace broken glass. In such instances, the hardened putty prevented easy removal of the cover plates. The use of a rubber or felt sealing strip is being contemplated and will be investigated. The house unit was found to withstand a high wind very satisfactorily, but the frames holding the cover glasses on the laboratory unit were blown off and demolished in the same wind storm. As a result, the type of construction originally used in the laboratory unit has been abandoned.

Snow removal constituted no problem, for the smoothness of the glass caused the snow to slide off the roof as soon as a thin layer next to the glass had melted. The $\frac{1}{4}$ inch mesh screen (hardware cloth) was considered adequate protection against hail damage in the summer and was used only in that season.

Ideally, a very thorough study of numerous fundamental problems should have preceded, or at least accompanied, the designing and testing of the whole solar collector. The limitations of time, personnel, and funds prevented this, however. Such studies as those directed toward learning the true character of the air flow between the heated plates, whether streamline, turbulent, or eddying because of thermal gradients, would be extremely valuable in establishing an optimum design. Measurements of air flow and temperature should be made by careful explorations across the cross section of the space between plates. Further theoretical development of the heat transfer relations should also be made and correlated with the experimental results.

It is hoped that some of these fundamental characteristics of the solar uni
can be investigated in a new project, along with the overall factors previousl
mentioned. With this more complete knowledge, it is felt that the most practi
design can be developed and the optimum operating conditions established.

VIII DISCUSSION OF RESULTS

The results of the study are divided into seven main sections: (1) a table showing results obtained with the outdoor laboratory unit, (2) graphs of these results plotted to show importance of certain variables, (3) a table showing house heating load carryable, calculated from laboratory outdoor unit data and house heating data, (4) a table showing size of heat storage system needed, (5) a table of actual fuel saving in the experimental house installation, (6) a table in which experimentally determined performance and the performance theoretically predicted are compared, and (7) a section of secondary results and correlations in which the results shown in table I and Figures 2 and 3 are plotted in numerous ways so as to show clearly the importance of certain operating variables.

TABLE OF RESULTS I

In Table of Results I, it is seen that runs 1 to 45 were all conducted with a spacing between plates of 1/4 inch. In these tests, the principal controlled variable was the rate of air flow which was varied between 32 cubic feet per minute and 369 cubic feet per minute, measured at 760 mm pressure and 70°F. Other conditions which varied because of weather and solar position were the inlet air temperature, the amount of incident radiation, and the angle at which the radiation struck the collector.

In runs 46 to 74, the spacing between plates was 1/2 inch, and air rates were varied between 33 and 354 cubic feet per minute.

Most of the runs were made in 1944 during August, September, and October, and a few were made in the winter and spring of 1945.

In the discussion immediately following, the various items in Table I are explained, and particularly significant results are discussed.

The overlap of 2/3 was employed because the results of tests on the small scale, indoor unit showed this to be the best arrangement. Plates 48 inches in length and blackened for 16 inches were employed simply because such a size was a convenient one with which to work, and not so small as to require an unduly large number of plates to fill the unit. The area of 209 square feet is actually the "black area", and does not include the area of end pieces and supports.

The spacing of 1/4 inch was found to be the optimum in the indoor tests and was utilized in the first runs on the outdoor unit. The spacing was later changed to 1/2 inch for another set of runs and the performance compared. As is seen in Figures 5 and 6, the 1/4 inch spacing proved to be superior, thus confirming the result of tests on the indoor unit.

Figures on time of sunrise and sunset were obtained from solar tables, and were not corrected for irregularities in the horizon. Such irregularities exist principally to the west of the unit and may cause the sunset to be as much as 30 minutes before the time shown.

The time at which the first direct sunlight strikes the collector is later than sunrise in the spring and summer, between March 23 and September 23 because the sun rises to the north of due east during these seasons. Since the unit slopes directly to the south, the time at which the first sun strikes the collector is when the sun has risen to a height such that it is due east of the unit. The same considerations apply in the afternoon, when the last sunlight strikes the collector as the sun crosses a position due west of the unit. In the fall and winter, the time at which the first and last sunlight strike the collector corresponds exactly to the time of sunrise and sunset, because the sun's position is always to the south of the unit during these seasons.

All the figures dealing with time of day are solar time, which at the longitude of Boulder, Colorado, corresponds roughly to Mountain Standard Time. Solar time varies from fifteen minutes ahead, to fifteen minutes behind, Mountain Standard Time, depending on the season of the year.

Statistics on sunrise, sunset, and time of first and last sunlight on the collector were used as a guide for establishing the duration of runs, and for an indication of the thermal lag or heat storage in the collector.

The cloud loss was calculated as shown in Appendix C, Methods of Calculation; the figures represent the percentage of the total possible radiation with completely clear sky which has been absorbed by clouds and not supplied to the collector on a given day. It is seen that the radiation lost because of cloudiness varied from zero to about 70 per cent. In most of the runs, however, the clouds reduced the solar input less than 20 per cent.

The cloudiness figures were used in determining whether any appreciable differences in over all heat recovery efficiency could be attributed to the extent of cloudiness. As shown in Figures 26 and 27, and as discussed below, very little difference in efficiency was found to be caused by clouds alone, even though they naturally reduced the overall heat supplied and recovered.

Since the calculation of heat balances required the determination of the heat lost by convection from the cover of the unit, and since the rate of air flow past the surface profoundly affects the rate of heat lost by convection, it was necessary to secure data on wind velocity. These data were not available at the site of the collector, so values from a station about 5 miles distant were used, after correction by the method shown in Appendix C. Although subject to some error, these data were sufficiently reliable to permit calculation of convection losses within the accuracy of the equations for determining convection coefficients. Moreover, those figures were needed only for the heat balances and are not involved in the efficiency calculations. For the computation of convection and radiation losses in the heat balances, it is necessary to know the temperature of the main body of air external to, but in the vicinity of the unit. It was found that this temperature was not the same as the temperature of the air entering the unit. Local weather bureau records showed that the atmospheric air temperature was always lower, sometimes as much as ten or twelve degrees, than the entering air temperature as measured by the thermocouple at the collector inlet. This difference can be attributed to the fact that the unit is surrounded by a large expanse of black roof which, when exposed to the sun, causes considerable heating of the adjacent air. Weather station records were, therefore, employed in the determination of the mean daily air temperatures for use in the calculation of convection and radiation losses.

Entrance air temperatures were measured as described previously and were used in the calculation of heat recovery. Since a stream of entering air was aspirated past the thermocouple junction at high velocity, the measured temperatures should be in error less than a degree. Variation of the temperature took place during the day, and the daily mean temperatures varied between 57° and 86°F through the main period of testing. Mean entrance air temperatures as low as 34°F were recorded in the winter tests.

Exit air temperatures were measured in the same manner as entrance air temperatures, but the effect of changing operating variables caused them to vary a great deal more. The exit air temperature is most markedly affected by solar heat input, air rate, and entrance air temperature. It is seen from the results that the daily mean exit air temperatures varied between 64° and 156°F., which values correspond to mean air temperature rises through the unit of 24° and 75° respectively. The maximum measured exit air temperature was 231°F. As is explained in greater detail below, the exit air temperature bears a direct relation to air rate and to heat recovery efficiency. Since no recycling of warm air to the unit was employed in these tests, entrance air temperatures were occasionally very low, especially in the winter. In the house installation, air is supplied to the unit from the house itself, and is therefore at an approximately constant temperature of 70°F. Thus, a temperature rise as small as 25° would allow delivery of air at 95° to the building.

The mean air temperature rises may appear surprisingly low, but it is to be realized that they are calculated simply by averaging the hourly exit temperatures and hourly entrance temperature, and taking the difference; whereas, a better picture, insofar as heat delivery is concerned, would be obtained by using a mean which is weighted according to the heat delivered. Thus, the temperature rises during the middle of the day, when high heat recoveries are being obtained, are particularly high, and of greater importence than those secured early and late in the day. In other words, for purposes of heat storage, it is more importent to have particularly high air temperatures during the period of high heat recovery, than it is to have a moderately high air temperature throughout the entire day. When the mean temperature rises are calculated in this manner (obtained most easily from the total heat recovered and the air rate), they are 10 to 20 degrees higher than those shown in Table I for most runs.

A second point in this connection is that the temperature rises during only the 4 or 5 hours in the middle of the day are considerably greater than either of the means mentioned above, and since roughly 2/3 of the daily heat recovery is in this period, the effective exit air temperature available to a heat storage unit is considerably above the averages shown.

A third consideration of the reported exit air temperatures indicates that these could be greatly increased by the use of a heat storage unit in a manner such that recycling of hot air from the collector to the storage unit, and back to the collector could be employed. With this system, most of the heat in the air leaving the collector is transferred to the storage bed, which results in heating of the bed and cooling of the air. As the bed's temperature increases, the temperature of the air leaving it also increases. If this warm air is circulated back to the collector, its temperature at the collector exit is raised considerably above that secured with the use of cold inlet air. The bed's temperature could be progressively increased by this recirculation method to the point at which the heat losses from the collector became equal to the solar input. It is expected that this point of maximum air temperature would be at least 250°F, and perhaps considerably higher.

Cover plate temperatures are probably subject to greater error than air temperatures because of radiation errors, and imperfect contact between the thermocouple strips and the glass. The cover temperatures were used only in calculating losses for the heat balances and probably are within the maximum probable error in the heat loss equations. It is felt that cover plate temperatures are generally within five degrees of the exact value. From the data, it is seen that the cover plate temperature is usually about 20° higher than the temperature of the surrounding air.

Temperatures of black surfaces were also measured so that the heat lost through the bottom of the unit could be calculated, and so that the exit air temperature and black plate temperature could be compared. The reliability of these figures is probably about the same as that of the cover plate temperatures, since the same method of measurement was employed. The data indicate that the average black plate temperature is 20° to 30° higher than the exit air temperature in most cases where reasonably high air rates were used. At lower rates the difference is greater and the plate temperature is higher. Thus, with a lower air rate, less heat is recovered, higher air temperature is obtained, and the unit operates at a higher temperature throughout. It is, therefore, apparent that the heat transfer coefficient is much lower at the low rates. Convection transfer, at least at the high rates, must therefore take place.

The air rate was varied over a wide range in order to determine its effect on exit air temperature and efficiency. The rate was controlled by adjusting a damper in the duct leading to the exhauster and measured either by the use of a calibrated rotameter or an orifice. Since equipment for continuously recording the air rate was not available, one or more measurements of air rate were made during each run by reading the rotameter or orifice manometer and the corresponding air temperature. Since the volumetric rate at the fan inlet remains constant throughout the day, even though the air temperature is steadily changing, the continuously recorded temperature at the fan inlet along with the above flow and temperature measurements were used in the calculation of air rates throughout the day. This method involved possible errors in the calibration of the meters, in temperature measurement, and in fluctuating voltage to the fan motor. It is felt that the maximum probable error in the mean air rates is less than 5 per cent.

More than a ten-fold variation in rate was secured, ranging from 32 to 369 cubic feet per minute. The total heat recovered during a day was calculated from the values of air flow rate, temperature rise through the unit, and the heat capacity of the air. To express the value on a basis of 1 square foot, the total was divided by 209, the area of the collector. Values were greatly affected by air rate, as further discussed below, and ranged up to approximately 1000 Btu per square foot of collector per day. The accuracy of these results is approximately the same as the flow rate accuracy and therefore within a maximum probable error of 5 per cent.

Gross heat input was calculated from the pyrheliometric data by integrating the recorder chart each hour and converting the results to engineering units. The recorder clock was set in accordance with the calculated solar time; the calibration of the pyrheliometer was directed by I. F. Hand of the U. S. Weather Bureau, Solar Radiation Section (23). Correction was also made for the seasonal variation in e.m.f. per unit of solar energy as caused by change in angle of solar incidence. By consideration of the possible sources of error in the above measurements and calibrations, it was concluded that the solar measurements could have a maximum probable error no greater than 2 per cent.

By the method described, the energy received per square foot of horizontal area was calculated. It was then necessary to convert these results, by a simple cosine relationship, to energy received per square foot of roof area sloping at a 27° angle.

Reflection losses were calculated from data on reflection from multiple glass surfaces (9) by the method outlined in the appendix. Since these losses vary with angle of incidence of sunlight, and since such angle varies throughout a day, hourly calculations were made and then totaled. The accuracy of these figures is high because of the reliability of the data and equations. The maximum probable error is thought to be less than one per cent.

Because diffuse radiation strikes the collector with a different effective angle than direct radiation, it was necessary to determine experimentally the portion of the total radiation which was diffuse. By shading the pyrheliometer with a small disc held several feet away, the diffuse radiation was easily measured. The presence of clouds causes a considerable change in the proportion of diffuse radiation, but because of the variability of this factor, and the small overall effect mentioned below, the mean angle of incidence during daylight hours was calculated on the assumption that all radiation was direct. Diffuse radiation is of most importance early in the morning and late in the afternoon when the amount of direct radiation may be small or even zero, whereas the diffuse radiation is a large portion of the total and strikes the collector at a much more favorable angle than the direct. However, the total reflection during the day is not appreciably affected by the portion of diffuse radiation because this portion is such a small fraction of the total, and its mean angle of incidence does not differ greatly from the mean angle for direct radiation. Further discussion of reflection losses may be found below, in the section devoted to heat balances.

Net heat input was calculated by subtracting the reflection losses from the gross heat input.. The principal use of the net heat input results was in the correlation of efficiency data. Since the reflection loss is affected by the mean angle at which the sun strikes the collector, and since this mean angle varies from day to day and season to season, the basing of performance on net heat input eliminates the effect of this seasonal variation when the effect of another variable, such as air rate, is being studied.

Gross efficiency is the percentage of the total radiation striking the collector which is actually recovered as sensible heat in the air stream. It is the primary index by which the performance of the collector is judged and should obviously be as high as possible. As explained in more detail below, it is most affected by air rate. In the tests shown in the Table of Results, gross efficiency varied from about 8 per cent at low air rates to approximately 40 per cent at high rates.

Net efficiencies are approximately 50 per cent at the high air rates. These values mean that about 50 per cent of the net heat input (gross input minus reflection) was recovered in the air stream. The usefulness of these figures is twofold: first, as described above in connection with net input, they allow a better comparison of performance on different days than do gross efficiencies, because the effect of change in solar position is minimized; secondly they represent more accurately the efficiency in terms of a fraction of the ultimate obtainable per-

formance with this collector. The latter consideration is valuable, because the reflected energy is not available for useful recovery, and therefore the unit could never have a gross efficiency of 100 per cent no matter what conditions prevailed. However, the unit could have a net efficiency approaching 100 per cent if the air rate were increased sufficiently. The net efficiency figures also indicate what overall performance (gross efficiency) might be obtained if surface reflection could be eliminated as by one of the surface treating processes for lenses and other types of glass. Further discussion of the factors affecting efficiency is presented in the explanation of the various plots.

In order to compare the total measured heat input with the measured useful output plus reflection, convection, conduction and radiation losses, a number of heat balances were calculated. In addition to showing the distribution of heat losses, these results give information on the reliability of the data and method of computation.

When the heat balances are studied it is seen that the unaccounted-for losse were frequently less than five per cent of the total losses (being therefore less than three per cent of total heat input). In view of the great amount of data ar calculations on which this result depends, and in consideration of the wide hourl variation in conditions frequently encountered and the resulting necessary approx mations, it is felt that the heat balances are highly satisfactory.

Several heat balances, in which wider deviations are noted, are perhaps not quite so reliable, but certain factors, particularly wind velocity, have such important bearing on particular losses that these results seem well within experi mental error. The distribution of losses is probably not so exact in these runs as in those with lower unaccounted-for losses, but the efficiency, which is basec n measured heat recovery, should be just as reliable as in the runs with less discrepancy in the heat balance.

There are two reasons for not calculating heat balances in all runs. In the first place, the overall recovery is the important practical item, whereas the distribution of losses is of considerably lesser importance in equipment performe Secondly, certain data, particularly cover plate temperature, necessary for heat balance calculations were not obtained in all runs.

In the table are shown the actual total daily heat losses by reflection from glass surfaces, convection from the cover glass into the surrounding air, conduct through the bottom and sides of the unit, and re-radiation from the warm cover pl into the surrounding air. The sum of these losses subtracted from the total (obtained by subtracting measured heat recovery from gross heat input) yields the unaccounted-for losses. Each of these losses was also computed as a percentage of the total.

It is seen from the heat balances that roughly one third of the loss is by reflection from the glass surfaces, another third is by convection from the cover plate, and the remainder is partially by conduction through the walls and floor of the unit and mainly by reradiation from the cover plates. The distribution of these losses, of course, varies considerably with such factors as wind velocit cover plate temperature, air flow rate through the unit, and solar intensity.

Reflection from multiple glass surfaces represents a large portion of the total losses, but there is little or nothing that can be done to reduce this, except by providing coated or etched glass (glass treated with certain agents to produce a coating of molecular thickness, to minimize surface reflection), a process as yet probably impractical for large glass surfaces. The glass could be made essentially non-reflective for certain wave lengths, but could not be made non-reflective to both the visible and infra-red radiation in the solar spectrum.

Reradiation and convection from the cover plate constitute together the greatest heat loss, and are dependent on the difference in temperature between the cover plate and the surrounding air. The convection loss also depends on the wind velocity.

The above temperature difference must exist if heat is being recovered, hence its magnitude should be maintained as low as is consistent with sufficiently high exit air temperatures. In other words, the higher the air rate through the unit, the lower will be the cover plate temperature and the resulting losses; and the higher will be the efficiency of heat recovery. If, on the other hand, high exit temperatures are desired, with no attention being paid to heat recovery efficiency, low air rates can be used, as for example in run 45, which result in high exit termperatures, high cover plate temperature, and high convection and reradiation losses.

High outside wind velocities increase the rate of heat loss by convection, but do not affect the re-radiation losses directly. A correction for wind velocity was made by the method shown in Appendix C, and should yield reasonably reliable results.

The equations which were used in calculating the various losses are based upon the standard methods used in heat transfer work, and are reasonably accurate. The calculation of the convection heat transfer coefficient is probably subject to the greatest error because of numerous factors, particularly wind velocity. The coefficient is markedly affected by wind velocity, and the estimation of velocities right at the glass surface is difficult. The accuracy of the heat transfer coefficient equation is also somewhat lower than could be desired. It is estimated that the figures on heat loss by convection have a maximum probable error of 10 per cent.

Radiation losses can be calculated reasonably exactly by the Stefan-Boltzman equation, provided that the cover plate temperature, air temperature, and surface emissivity are accurately known. In some runs, reliable cover temperatures were available, and it is believed that these were within one or two degrees of the exact surface temperatures. As explained previously, Weather Bureau data were used for calculations of the temperatures of the surrounding air. It is felt that the radiation losses have a maximum probable error less than five per cent.

Knowledge of reflection losses is obtained entirely by calculation of reflection from multiple glass surfaces. These figures should be even more exact than those described immediately above, because the only complicating factors are dirt on the plates, and change in angle of solar incidence. Dirt on the plates has been shown (9) to have negligible effect on the reflected energy; and small changes in angle of incidence, in the ranges involved, cause changes relatively small in the portion reflected.

Conduction losses through the floor and walls are small, and calculation of their values has been made by simple and fairly exact methods. Errors would not exceed those mentioned above.

The unaccounted-for losses do not represent other types of losses, but rathe additional losses of one or more of the above types. In two or three instances, negative unaccounted-for losses indicate that one or more of the other reported losses is no doubt high.

In general, the heat balances show a remarkable correlation of the data and lend strong support to conclusions based thereon. They also show how the various losses are distributed, and which ones should first be investigated further in an effort to improve efficiencies.

PRIMARY GRAPHICAL RESULTS

In the numerous graphs of results, several variables have been correlated to show more clearly than does the table how the performance of the unit is affected by changes in its arrangement and operation. These graphs have been divided into two separate main sections, the first of which may be considered to contain the principal basic plots, and the second to contain the secondary graphical results and numerous correlations. The purpose in this division is to show clearly the overall performance of the solar unit by a few simple graphical analyses, and then to show the specific effects on exit air temperature and efficiency of varying air flow rate, entrance air temperature, plate spacing, heat input, cloudiness, and season of the year.

The graphs in the first section may be further classified as follows:

1) A sample graph showing how the performance of the unit varies with the hours of the day. (Figure 4)
2) Graphs showing relation between air rate and efficiency for different plate spacings. (Figures 5 and 6)
3) A graph showing the relation between angle of collector tilt and the winter radiation collected. (Figure 7)

The graphs in group 2 probably show in simplest form the general overall performance of the equipment. Figure 4 is included to show how the heat recovery and air temperature vary throughout a clear day, and is an example of the sort of data which were used in the calculation of final results shown in the tables and other graphs. Figure 7, on the other hand, is a graph of the relative effectiveness of collectors at different tilts, and would therefore be useful in the design of solar heating units.

By reference specifically to Figure 4, it is seen that the heat input rises rapidly to a maximum at noon, then drops off in the afternoon nearly to zero at sunset. On cloudy days, the curve is not perfectly regular, of course. The heat recovery curve is of a shape similar to the input curve, but lower and displaced somewhat to the right. Its lower height is due to the heat losses previously discussed, and its displacement is caused by the thermal lag in the unit. In other words, the heat capacity of the glass and other components of the unit is such that approximately one hour is required to heat the equipment to operating temperature. Thus, trends in the heat input curve are not evidenced in the output curve until about one hour has elapsed. In the morning, this effect causes a delay in useful heat delivery, whereas in the afternoon, a prolonged heat delivery is noted. This extra heat delivered in the afternoon hours, some even after sundown, can be looked upon as morning heat stored in the unit.

The areas under the heat input and output curves correspond to the total respective heats for the day, and the ratio of the two areas can be used to obtain the heat recovery efficiency, 37.4 per cent in the case shown.

Temperature variation throughout the day is also shown in Figure 4. It is seen that the exit air temperature curve has a shape and position almost identical to those of the heat recovery curve. In other words, the same lag exists in the attainment of elevated air temperature as in securing heat recovery, because the one quantity depends on the other. In the case shown, the temperature difference

between entering and exit air was at a maximum of 72 degrees at 1 p.m., solar time. Moreover, exit temperatures were at least 20 degrees higher than entrance temperatures for more than eight hours during this particular day. The entrance air temperatures showed a natural rise and fall during the day.

In Figures 5 and 6, gross and net efficiencies are plotted against air rate, for two different plate spacings. For Figure 6, the calculated reflection losses were deducted from the total heat input, and the resulting net heat input figures were used in computing net efficiencies. Gross efficiency, plotted in Figure 5, is the actual heat recovered divided by the total heat input.

In these plots, as in others, there is very little difference between the trends in the net and gross efficiencies. This means that the reflection losses from the unit as a percentage of input were sufficiently constant throughout the several months of recorded runs that the net efficiencies are simply proportionately greater than the gross efficiencies. The seasonal change in the sun's position caused practically no difference in the percentage reflection.

It is seen in Figures 5 and 6 first that the 1/4 inch spacing yields decidedly greater efficiencies than the 1/2 inch spacing. This is in complete agreement with the results of tests with the small scale indoor unit. The superior performance of the unit with 1/4 inch spacing is no doubt due to the decreased eddying of the air passing between plates and the resulting lower convection heat transfer upwards from plate to plate.

A second important conclusion drawn from Figures 5 and 6 is that efficiencies are low at low air rates, rise nearly linearly as air rates are increased to a moderate value, and then rise more slowly as air rates are increased to a comparatively high figure. It is also seen that with 1/4 inch spacing, efficiencies increase only slightly above 37 per cent gross and 50 per cent net, as air rates are greatly increased. This fact points toward the conclusion that with the apparatus tested, there is practically no advantage in going to excessively high air rates in order to secure slightly higher efficiencies. That this point is particularly important is realized when it is noted that the maximum exit air temperature decreases almost in inverse proportion to the air rate.

It is of course true that if the air rate were increased to a sufficient degree, the net efficiency would very nearly reach 100 per cent. The net efficiency curves are therefore not approaching maxima near the values shown in Figure 6. That net heat recovery efficiency should approach 100 per cent at high air rates is obvious because the higher the air rate, the lower the temperatures in the unit, and hence, the lower the reradiation, convection, and conduction losses. These losses would approach zero at sufficiently high air rates. It is felt that the apparent slowness of the approach to 100 per cent net efficiency is caused by the increased air turbulence at high rates, and the resulting increased ease of heat transfer by convection from plate to plate. Thus, as air rate is increased, although the temperature in the unit is reduced, the convection heat transfer from plate to plate is facilitated, the cover temperature does not decrease greatly and the losses remain relatively high. Hence, efficiency does not increase markedly even with large increases in air rate.

A third fact apparent in Figures 5 and 6 is that cloudiness has no noticeable effect on efficiency. Points differentiated according to clarity of sky show no noticeable trend, and the curves were therefore drawn through the whole group of points. Another plot of this effect is presented in Figures 26 and 27.

In Figure 7, the relation between total radiation collected and change in angle of tilt of collector is presented. For a given collector tilt, readings on the vertical scale indicate the ratio of the total radiation which would be collected during the period October 1, 1944 to June 1, 1945 to the radiation which would be collected during the same period by a collector of the same area, but tilted at 40 degrees from the horizontal. The graph was constructed from results of two complete heat recovery calculations. In the first of these, solar energy received each hour per unit of horizontal area, as recorded by the pyrheliometer equipment, was totalled each day. This result was then converted to heat recovered by a roof unit tilted 40 degrees to the south from the horizontal and collecting heat with an overall daily net efficiency of 45 per cent. The total radiation thus collectable by the 463 sq. ft. unit over the winter period would be 64,630,000 Btu. A similar calculation was made for the case of a roof sloping 27 degrees from the horizontal, and the ratio of the total heat recovered by the 27 degree roof to that by the 40 degree roof was determined. It was also found by calculations based on the relation between solar position and season (22) that at a latitude of 40 degrees the angle of tilt at which the maximum radiation can be collected during this period is 43 degrees. This result is obtained as shown in Appendix C by graphically integrating an equation in which the total radiation received by a tilted collector during the winter season is equated to a function involving latitude, angle of collector tilt, solar declination, and atmospheric transmissivity. By repeating the calculation for several collector tilts, the one for which the maximum radiation is received is determined. Although the final result is based on the solar position at noon of each day, it is valid for the entire day, because optimum exposure at noon is accompanied by optimum exposure all day.

Even though the calculation of the optimum angle of tilt in Figure 7 is based on calculations of solar position whereas the 27 degree and 40 degree points are based on actual radiation received, the correlation appears very good. This relation indicates in turn that such factors as cloudiness reduced the monthly radiations symmetrically throughout the period between October and May, and did not predominate heavily during any month.

The points at 46 degrees and 59 degrees tilts are obtained by analogy and involve the same ratio of radiation collected as do the 40 degree and 27 degree points. Such is the case because the collectors are merely tilted by the same number of degrees beyond the optimum, hence they receive radiation at the same angle but simply oblique in the opposite direction.

The remaining two points which are used to construct the curve are located in accordance with the fact that the mean angle of 47 degrees (90 degrees minus 43 degrees) which the sun's rays make with the horizontal through these months would cause the rays to be parallel to the surface of a unit sloped 47 degrees to the north. Thus, if a collector's surface is tilted 47 degrees to the north, and if the sun were stationary at its average noon winter position, the sun's rays would simply graze the surface of the unit and do not enter it. Hence the ratio of radiation recovered would be zero. Actually, some small amount of radiation would be collected because the 47 degree solar angle of elevation is simply a mean, and there would be periods in the fall and spring in which some radiation would be

collected. However, this effect would be small, and has not been computed in this analysis. Similarly, if the unit were tilted to the south, considerably beyond the vertical position so that its cover surface faced down at an angle of 133 degrees, essentially zero collection would result.

Although the seven points described above do not constitute a complete and precisely accurate set of results, the curve drawn through them has considerable meaning and value. It is seen that at 40 degrees latitude, a tilt of 30 degrees is 94 per cent as effective as one tilted at the optimum 43 degrees, and a unit tilted at 40 degrees is only about 1 per cent less effective than the optimum. Thus, the effectiveness of tilting the units to different degrees, as for example, to correspond to the roof slopes of different houses, can be readily observed in Figure 7. The graph clearly shows the importance of tilting the collector, because if a horizontal position were used at the 40 degree latitude, the energy collected during October through May would be less than 70 per cent of that collected by a unit of equal area tilted at 43 degrees. The graph can also be used to calculate directly the relative areas of units collecting the same radiation, but tilted at different angles. Thus, a collector tilted at 27 degrees should be 8 per cent (100-92) larger than one tilted at 40 degrees, if the same total radiation is to be collected.

It is to be recognized that a tilt of 43 degrees is the optimum for heat <u>collection</u>, but not necessarily for heat <u>use</u> in a house. In the fall and spring, a large portion of the collected heat would be discarded, because the heating load is light in those seasons. Hence, the optimum tilt would be somewhat steeper than 43 degrees in order to favor heat collection in the winter months when essentially all the collected heat is needed in the house. In order to calculate such an optimum, the fraction of the collected heat which would actually be used in the experimental house each month was obtained from Table II. Values for a 40 degree tilt and one day storage were employed. By using these fractions as multipliers in the previously mentioned radiation equation, integration of the function for various tilts could be performed. It was found that with collector size and heating requirements analogous to those in the experimental house, a collector tilted at 47 degrees would carry a greater portion of the annual heating load than would a collector at any other tilt, provided that heat could be stored for a one day period. Theoretically, the values for monthly fraction of collected heat used by the house should be recalculated for a 47 degree tilt rather than 40 degrees, but the effect on the final result would be negligible. The variation in solar radiation caused by clouds has not been considered, but the effect of such variation is relatively small. The use of collectors larger than the one on the experimental house, relative to the heating requirements would require tilts somewhat greater than 47 degrees if the optimum were to be maintained. Smaller collectors, or larger heating requirements, would entail the use of an angle of optimum tilt less than 47 degrees, but not less than 43 degrees. In general, therefore, a house at 40 degree latitude, in which approximately the same monthly fractions of collected energy are actually used as in the experimental house, should be provided with a collector tilted at 47 degrees if the collector area is to be at a minimum.

At latitudes other than 40 degrees, Figure 7 would be somewhat different, but roughly, the optimum tilt for heat <u>collection</u> during this winter period could be found by adding approximately 3 degrees to the latitude. Thus, at latitude 30 degrees, the collector should be tilted about 33 degrees from the horizontal to

receive the maximum radiation in this period. The 3 degree figure is the approximate mean declination of the sun during this period, that is, its average position is 3 degrees below the equator. To collect and utilize the solar heat in a house having a heating requirement and collector area similar to the experimental house and having one-day heat storage capacity, the collector should ideally be tilted at an angle equal to the latitude plus approximately seven degrees.

If the collector were used primarily to supply energy to an air cooling unit operating in the summer, it should, of course, be oriented more favorably for summer exposure. Tilts of 20 degrees to 25 degrees would probably be near the optimum during this season.

TABLE OF RESULTS II

Table of Results II contains the calculated values of the percent of the tot heating load carried by the solar unit from October 1944 through May 1945. The data are given for the conditions that (1) no heat storage is used, and that (2) excess heat is stored for one, two, or three days. Data for both a 27° and a 40° tilt of the unit from the horizontal are given.

The calculations of the data for operation with no storage involved the use of degree-day figures for the daylight hours only. These figures were determin from daily temperature charts available from the Geology Department at the Univer sity. The heat required per degree day was originally calculated on a twenty-fo hour basis, but since sunlight adds to the house heat supply, there would be les heat required in the hours of daylight per degree-day than would be required during the night. In calculating the heat required during the daylight hours, this difference was neglected because of the complexities involved in determinin the correct values. The calculated heat requirement during the daylight hours i therefore, somewhat higher than actually would be needed.

The heat storage calculations were made on the assumption that there was storage capacity sufficient to hold all of the excess heat collected for a certa period of time, in this case one, two, or three days. The calculations could have been made on the assumption that storage capacity was provided that would hold a certain maximum quantity of heat, but this method would require a longer set of calculations. By examination of the results obtained by the first method of calculation, the necessary capacity of the three sizes of storage units could easily be determined. These storage unit sizes are shown in Table of Results II. All of the original calculations were made with the further assumption that ther was no loss from the storage unit until the one, two, or three day period had elapsed. At the end of the storage period, all heat which had not been used was assumed to be lost. This assumption is not strictly true, as the heat is gradu- ally dissipated, but it greatly simplifies the calculations without introducing serious error.

The data show the advantage of incorporating a storage unit into the solar heating system. For the 27° tilt, the addition of a storage unit capable of sto ing all the excess heat for one day, would increase the percent heating load carried from 34.0 to 54.8, but increasing the storage capacity the amount necess to store heat for 2 or 3 days would increase the percent heating load only to 58.2 and 60.1 respectively. These data show, therefore, that the greatest ad- vantage is obtained with a small storage unit capable of storing heat for one day, and that the use of larger units may be impractical. The final design, how- ever, will depend on an economic balance which will show if the added cost of la storage facilities will be greater or smaller than the value of the additional heat saved.

It is interesting to note that in October more heat was collected in the sol unit than was required to heat the house, but when no storage was provided, only 30% of the heating load was carried. The reason for this anomaly is that the ex- cess heat was collected during the day and discarded, and was, therefore, not available when heat was needed in the evenings and early mornings. If storage

facilities had been provided, this waste heat could have been saved and used when needed. During the remaining months, the heat received was not enough to carry the entire heating load, but, if desired, more heat could be collected by employing a solar heat trap of greater surface area. An economic balance of cost of solar unit versus fuel cost would be of assistance in determining the optimum size of solar unit to use.

The angle the collecter makes with the horizontal is shown to have a definite influence on the operating characteristics of the heating system. To receive the maximum amount of solar radiation at any time, the unit should be normal to the sun's rays. Since the unit is in a fixed position, the angle of incidence of the sun on the unit must be chosen to give the maximum amount of radiation over the period considered. For the period shown in the table, this angle has been found to be about 47 degrees, when the house heating needs are considered and one-day storage is provided. The table shows that a 5% increase in heat recovered could be expected if the angle of tilt were increased from 27° to 40°. Forty degrees roof tilt was chosen as about the maximum practical angle.

TABLE OF RESULTS III

In Table of Results III are shown the necessary sizes for heat storage units packed with a material having a heat capacity of 0.2 Btu/lb.°F., such as brick-work, coke, clay, glass, granite, or stone. A maximum available temperature dro of 130°F. was assumed. Thus, the unit may be considered to have a maximum quanti of heat stored when at a temperature of 210°, and no useful heat when at 80°F. If the exhaust air from the storage unit were to be recycled back to the solar unit intake, the maximum useful temperature drop might be considerably increased As reported in the literature, water or some other liquid might also be used as a heat storage medium. Since much more elaborate equipment would be required fo the use of a liquid storage medium, it would probably be cheaper to store the heat in a chamber of loose solids. For this reason, heat storage in a liquid ha not been considered in this calculation.

The amount of excess heat recovered each day by the solar unit and stored for subsequent use was first calculated. At infrequent intervals an abnormally large amount of excess heat was recovered. The figures in column 5 of Table III are these maximum quantities of stored heat, regardless of whether or not the heat is needed. It would seem unnecessary to design a storage unit with capacit sufficient to store this abnormal amount of heat if satisfactory results could be obtained by using a unit which would store a normal maximum or average high quantity. Columns 4 and 7 of the table show clearly that the difference in storage sizes would effect a difference in heating load carriable of only 1 to 2 per cent, whereas the larger unit would require at least 33 per cent more material.

The difference in storage unit size required for one, two, and three day heat storage is also shown in Table of Results III. The final design of the storage unit would be dictated by an economic balance, but superficial examinati of the data leads one to believe that the economical unit would be designed to store heat for one day. To store heat for three days, a storage unit nearly thr times larger would be required, and only 4 to 6 per cent increase in heating loa carriable would be realized. The approximate volume of a unit designed to store the normal maximum of excess heat for one day would be five cubic yards.

TABLE OF RESULTS IV

In table of Results IV A, B, and C, actual operating data on the experimental house heating system are presented. The standard heating plant in the experimental house comprises a furnace in which natural gas is used as fuel, a fan and motor for delivery of hot air to the rooms of the house, and thermostatic controls for temperature adjustment. Into this system was incorporated a solar heat collector and the necessary ducts, dampers, and additional controls for automatic operation. Fuel saved in the experimental house after the solar unit was installed was determined by comparing the gas consumption in two identical houses over the same period (Table IV A) and by comparing fuel requirements in the solar heated house before and after the solar installation (Table IV B).

In Table of Results IV A, the gas consumption of the solar heated house is compared with that of an identical house not equipped with a solar heating system. The second house has the same floor plan and volume as the experimental house, and is situated adjacent to it. During the year preceding the installation of the solar heating unit, the average gas consumption in the test house was 25 per cent greater than in the identical house next door. The mode of living of the respective occupants and the temperature level maintained in the two houses combined to cause this difference in heating requirements. During the period shown in the table, the temperature of the test house was maintained about 5 degrees higher than it had been the previous year. This increase in temperature would increase the heating load, and hence the gas consumption, of the system. The amount of this increase was estimated by the Public Service Company of Colorado to be 3.1 per cent for each degree increase in temperature (19). For the 5 degree increase in temperature, the gas consumption should be increased 15.5 per cent. The gas consumption for the test house, if the solar unit were not used, should then be 25 per cent plus 15.5 per cent or 40.5 per cent greater than that for the identical house.

A number of errors are inherent in the calculation of the results shown in the table. The figure used in the comparison with the identical house was only an approximation since (1) the heating load increase of 25 per cent was based on data for only one winter, and during that time it varied considerably; (2) the five degree increase in temperature maintained was an assumed average for the entire heating season, and probably somewhat inaccurate; and (3) the gas consumptio increase of 3.1 per cent per degree was an average for many houses and may not have been exact for this particular one. Other errors which might have been of consequence were possible differences in the gas consumed in the two automatic hot water heaters and in the kitchen range of the identical house during the two years of comparison.

The last column in Table of Results IV A shows the per cent heating load carriable as determined from the amount of gas saved (Column 4). It would seem that in the spring and fall. when a large amount of heat was collected, the per cent heating load carried would have been greater than in the winter when smaller quantities of heat were recovered. The table shows that in reality the reverse was true. In the spring and fall, most of the heating was required at night and a great amount of heat was thrown away during the daylight hours, but in the colder months the per cent of the total heating required during the daylight hours increased and more of the collected heat was utilized. It is, therefore, seen that in the absence of heat storage facilities a greater per cent of the heating load was carried in the winter months.

The discrepancy between the per cent heating load carriable shown in Table II and that shown in Table IV A may be attributed to factors other than those mentioned above. The figures in Table II were based on an air rate of 180 CFM and the corresponding net efficiency of 45 per cent obtained with the laboratory unit. Since the area of the laboratory unit was 209 sq. ft. and that of the house unit was 463 sq. ft., an air rate of 400 CFM should have been maintained in the house unit, if it were to operate at an efficiency of 45 per cent. From experimental measurements made at noon, an air rate in the house unit of only 188 CFM and an efficiency of 12.8 per cent were calculated. If the galvanized iron ducts in the house unit were insulated, the shading effect of the chimney eliminated, and the broken glass replaced, the efficiency should be about 29 per cent. This value can be read from Figure 6 at a laboratory unit air rate of 85 CFM, which corresponds to the house unit air rate of 188 CFM. The efficiency of the house unit, 12.8 per cent, may be somewhat in error as it was assumed that the efficiency at noon was approximately the same as the efficiency for the whole day. This approximation was found to be reasonably valid by examining the laboratory unit data. An additional difference lies in the fact that the figures for heating load carriable shown in Table II are based on only one year and would, of course, change from one year to the next.

Results of calculating house unit performance by use of degree-day heating requirements, instead of by direct comparison with the adjacent house, are shown in Table IV B. Since the number of degree-days of heating is only one of many heating load variables, such as wind velocity, sunshine, mode of living, and thermostat settings, the heating load carried shown in column 9 is subject to considerable error. The fact that the heating load carriable in September, October, and November is recorded in the table as zero, is a good indication that the method of calculation is not reliable, because it was known that some of the required heat was furnished by the solar unit.

The actual operating data for the house heating system, as determined experimentally, are shown in Table IV C. The predicted heating load carriable is based on the assumption that all the heat collected at the low efficiency of 12.8 per cent was used during the day and that lack of storage was not a disadvantage.

Since the first method of calculation, that of comparison with the identical house, and the method involving the 12.8 per cent efficiency figure are in the same range, and since the degree-day method is known to be seriously in error, the most probable average per cent heating load carried by the present house solar unit is about 20 per cent. Though this figure is low in comparison with the 34 per cent heating load carriable shown in Table II, it could be increased simply by supplying the correct air flow and insulating all the hot air ducts.

TABLE OF RESULTS V

Comparison of Actual and Theoretical Results

In the report of Miller (10), theoretical consideration of the operation of a solar heating unit was made, and quantitative results of its operation were predicted. In the calculations, it was assumed that (1) no heat transfer took place by convection from plate to plate, (2) no heat was lost to the surroundings by convection from the cover plate, (3) the radiation loss from the cover plate was to a space temperature of absolute zero, (4) no heat losses from the sides and bottom of the unit took place, and (5) certain simplifications of the rigorous mathematical solution could be made without introducing large errors in the results.

In order to make a comparison between experimental and theoretical results, it was necessary to use Miller's method in the calculation of a set of results based on a specific outdoor unit run. Since convection from the top plate was known to take place, the amount of heat lost in this manner was determined as shown in Appendix C. The reradiation from the top plate was assumed to take place to the outdoor air temperature, since this has been found to be a better approximation than the one proposed by Miller, which states that radiation takes place to a space temperature of absolute zero (9). Miller's assumption (4) in which he stated that there was no loss of heat from the sides and bottom of the unit is, of course, not rigorous. In the present calculations, however, it was unnecessary to consider this loss because it would take place from the heated air after the air left the black plate, whereas the air temperature was measured at the plate end before any heat loss took place.

Outdoor unit run 0-42 was chosen for the comparison because it was made on a clear day for which all necessary heat balance data were available. The calculations are made for the hour ending at solar noon. Average values for the hour were used in every case.

Miller's analysis was based on a unit with no cover plate. To make a rigorous comparison of the experimental and theoretical heat traps, it would be necessary to determine the heat interchange between the top plates and the cover plate, and to determine the amount of heat picked up from both plates by the entering air stream. The complexities involved in making the above calculations make it impractical to include them in the present comparison, hence it was decided to make two theoretical calculations in order to obtain a basis for comparing the experimental results. The first analysis was made on a theoretical unit with three plates in a stack, which is essentially a blackened plate insulated with two clear plates (2/3 overlap); and the second analysis was made on a unit with four plates in a stack (3/4 overlap). In the first analysis the unit had the same number of ideal heat transfer surfaces as the experimental unit, but less radiation insulation and reflection. In the second analysis the unit had more heat transfer surfaces than the experimental unit, but the same amount of radiation insulation and reflection. Since all calculations were based on the same top plate temperature, convection and reradiation losses were the same.

The results of the calculations are shown in Table of Results V. If the theoretical analysis were entirely valid, the experimental temperatures should have fallen between the two calculated temperatures in each case. This generalization is seen when it is realized that the temperatures in the experimental unit should have been below those in the theoretical 4 plate unit because there was actually one less ideal heat transfer air film but the same amount of reflection and reradiation, and that experimental temperatures should have been higher than those in the three plate theoretical unit because there was actually one more glass plate opaque to reradiation to the sky; this effect may have been partially counter-balanced, however, by the additional reflection caused by the extra glass.

Since the experimental temperatures did not fall between the two sets of theoretical temperatures, one correction was thought to be in order. The unaccounted for loss of 24 Btu in the experimental unit should have been included in the heat balances in either of two ways. The 24 Btu might have been added to the losses in the theoretical calculation and a new calculation of temperatures made, or this loss might have been added to the heat recovery of the experimental unit. Though neither of these methods is strictly accurate, either could have been employed to make possible a better comparison of the experimental and theoretical units. The latter method was chosen. After adding the 24 Btu to the 158 Btu heat recovery in the experimental unit, the revised heat recovery and efficiency were calculated to be 182 Btu and 55.4% respectively. These values are necessarily the same as those obtained in the 4 plate theoretical calculation since both units had the same number of plates and the same top plate temperature. Under these revised conditions, the temperature of the air leaving the black plate was calculated to be 201°F. This value is higher than the experimentally determined value of 186°F., but it is still below either of the theoretical values. If the 24 Btu had been added to the losses in the theoretical calculation, the theoretical temperatures would have been lowered somewhat, but probably not enough to alter the comparison greatly.

The difference between the experimental results and the revised theoretical results may be attributed largely to the assumption that no heat transfer took place by convection from plate to plate. If the air flow were strictly laminar, no convection would take place, but the very large temperature gradient across the air film is a good indication that laminar flow did not exist.

It should be mentioned that the difference in air rates in the experimental and theoretical results in Table V does not constitute a reason for the observed disagreement of temperatures. All calculations are based on equal top plate temperatures, and any change in air rates from those listed would change this top plate temperature and hence change the reradiation loss. If the theoretical analysis were precise, therefore, the experimental air temperature and rate would be identical with those calculated, provided that the same top plate temperature prevailed.

In tests on the experimental unit, two inaccuracies which may be partially responsible for the discrepancy in the results just noted are (1) inaccuracies in air temperature measurement, and (2) the time lag from radiation received and heat recovered. The air temperatures were measured by high-velocity thermocouples, but may be slightly affected by the usual errors encountered in thermocouple pyrometry. The solar trap necessarily shows a considerable time lag between the moment the solar rays strike the unit and the time it attains a temperature corresponding to this input. This time lag is due to heat storage in the glass plates

and in the side walls and bottom of the unit. It has been noticed that the air traversing the unit would show its maximum temperature rise about one hour after the time of maximum solar input (solar noon), and it has therefore been concluded that the time lag is approximately one hour. Because of this lag, the air temperatures and surface temperatures are undoubtedly not in equilibrium with a particular incident solar radiation, nor does this equilibrium ever exist with a changing sun. The assumption of equilibrium is therefore, no doubt, in error. A better correlation might be obtained in the calculation of theoretical performance by using the average incident solar radiation for the hour preceding the temperature readings. Since the unaccounted for loss includes heat stored in the one-hour period, the above method would, however, be an inclusion of a loss correction for which has already been made. Therefore, correction for the time lag has not been separately made.

SECONDARY GRAPHICAL RESULTS AND CORRELATIONS

Figures 8 to 31 comprise the secondary results and correlations obtained with the laboratory unit and may be further classified as follows:

1) Graphs showing how efficiency, air rate, and air temperature rise are related at constant heat inputs (Figures 8 to 13)

2) Graphs showing how efficiency, entrance air temperature, heat input, temperature rise, cloudiness, and angle of solar declination are related at constant air rates (Figures 14 to 31).

The above graphs are based on the data and results presented in Table I. They are used to show in detail the specific effects of changes in certain variables and are therefore of considerable value in explaining certain deviations from the average noticed in the general table of results (TableI) and in Figures 5 and 6. They also show clearly the particular effects of changing one operation variable while holding the others reasonably constant. In several cases, graphs are presented which because of similarity to each other, show essentially the same facts and from which the same conclusions are drawn. These similar graphs have all been included however, so that a complete picture of the results could be presented.

In Figures 8 and 9, air temperature rise is plotted against air rates at the two plate spacings and at several constant daily heat inputs. The reason for correlating these variables at constant net heat input is that air rate and temperature rise could not show a simple interdependence under all conditions of small and large heat inputs, whereas at a certain heat input, temperature rise should vary systematically with change in rate.

Comparison of the two figures shows that air temperatures are higher with the 1/4 inch spacing than with the 1/2 inch spacing at corresponding air rates and heat inputs. Each figure shows also that the temperature rise is greater with the larger heat inputs, and that at constant input, the air temperature rise decreases linearly with increase in air rate. The latter effect is to be expected qualitatively, and is of value in determining quantitatively what air rates should be utilized in order to obtain particular daily average air temperature rises. Maximum average daily rises of about 70 degrees F can be secured at low air rates, and average rises of 30 degrees to 40 degrees can be secured at high rates, with resulting high efficiency.

In Figures 10 to 13 correlation of efficiencies with temperature rise at constant net heat input is made. As mentioned previously, the same trends in net efficiency and gross efficiency are noted, and Figures 12 and 13 are therefore very similar to Figures 10 and 11.

It is seen in Figures 10 and 11 that better performance is obtained with 1/4 inch spacing than with 1/2 inch. In other words, at a fixed heat input and a given efficiency, a better temperature rise is secured with 1/4 inch spacing; or for a given temperature rise better efficiency is observed.

In each of the four graphs, the lines are relatively close together, and there is considerable scattering of the points. It thus appears that the relationship between daily average air temperature rise and efficiency is not markedly affected by small variation in heat input. It would be expected, however, that with a greater difference in solar input, wider variation in efficiency would be observed. Three such points are shown in Figures 10 and 12 at net heat inputs of 800 Btu/ft^2. If the air temperature rise is to be the same with this lower input as with the higher inputs, the air rate must be considerably lower. For the same temperature rise, approximately the same cover temperatures exist and hence the same rate of heat loss. These losses therefore constitute a larger percentage of the input and cause a lower efficiency. This poorer performance is observed in Figures 10 and 12 at the net heat input of 800 Btu/sq.ft. There is not enough variation in the other inputs to cause a marked difference in the relationship between efficiency and temperature rise.

It is, of course, seen that a high efficiencies (corresponding to high air rates) the temperature rise is comparatively low, and that as air rate is decreased, the temperature rise increases, losses increase, and efficiency decreases. These plots can be useful in establishing the conditions for operation of a solar heating unit because they show clearly the balancing of the two most important variables temperature rise and efficiency. Thus, it can be immediately seen from Figure 10 that a 50 degree temperature rise can be obtained with about 26 per cent gross efficiency or a 40 degree rise with 30 per cent to 35 per cent efficiency. The air rates corresponding to these conditions can then be ascertained from Figure 5.

Figures 14 to 17 show the relation between entrance air temperature and efficiency. Figures 14 and 15 show how gross efficiency varies with entrance air temperature at the two spacings, and Figures 16 and 17 show the variation in net efficiency. These factors have been correlated at several constant air rates in order to eliminate the effects of this variable.

It is seen in all four figures that a change in entrance air temperature has only a slight effect on efficiency, and that the efficiency decreases slightly as entrance air temperature is raised. This effect is explained by the fact that with higher entrance air temperatures, the unit, as a whole, operates at somewhat higher temperatures, which causes higher losses and slightly lower efficiencies. In practical use, some advantage is gained by this effect, because with the low entrance air temperatures encountered in the winter, efficiencies are at their highest values.

In Figures 18 to 25, the relation between heat input and efficiencies is presented. As in the preceding plots, all comparisons are made at constant air rates. Plots shown are gross input vs gross efficiency, net input vs. gross efficiency, gross input vs net efficiency, and net input vs net efficiency at the two different spacings.

All the graphs show essentially the same general trends. In Figure 18, for example, it is seen that as heat input is increased, while air rate is held constant efficiency decreases slightly. The doubling of heat input appears to decrease the efficiency by only five per cent. This change is due to the higher temperatures and losses at the higher heat inputs and to the more rapid increase in losses than in heat input. That is, if heat input is doubled, the losses are more than doubled thus slightly reducing the recovery efficiency.

In Figure 19, the changes are again seen to be slight. The lines in the case of these runs with 1/2 inch plate spacing, however, have a slope opposite to that of the lines obtained in the runs with 1/4 inch spaced plates. Very little significance is attached to this difference because the points are so few and scattered that a small error in one or two of them could easily change the slopes of the lines to correspond to those in Figure 18. As a matter of fact, there is probably equal justification for drawing the lines in Figure 19 with a slight negative slope. It can therefore be said, in general, that with 1/4 inch and 1/2 inch spacing, efficiency is not appreciably affected by change in solar heat input at constant air rates, and if there is a slight effect, it is a decrease in efficiency as heat input is increased.

The practical value of this fact is that cloudy skies and hazy atmosphere do not appreciably lower the <u>efficiency</u> of heat recovery, although the total input, and hence the output, are considerably decreased. This point is further discussed below in connection with Figures 26 to 29.

Figures 20 to 25 show the identical facts as 18 and 19 described above and merely involve the plotting of <u>net</u> heat inputs and <u>net</u> efficiencies rather than the <u>totals</u> used in Figures 18 and 19. As explained previously, trends in net heat effects are seen to parallel those of gross heat effects.

The effect of cloudiness on efficiency is shown in Figures 26 and 27. It is seen that there is practically no change in the efficiency, even when cloudiness reduces the normal radiation by one-half. There appears to be a very slight increase in efficiency at high cloudiness, possibly because of lower exit air temperatures and resultant disproportionately low heat losses. There may also be a slight improvement in the mean angle of incidence of solar radiation, causing slightly lower reflection losses. The constancy of efficiency is advantageous in the practical operation of solar units, because the unit works as well in cloudy weather as in clear, but of course less heat is delivered because of lower input.

In contrast to the slight effect of cloudiness on efficiency, Figures 28 and 29 show the marked effect of cloudiness on air temperature rise at constant air rates. As would be expected, the air temperature rise is not as great when radiant input is reduced by clouds, provided that the air rate is unchanged. With a lower heat input, a lower recovery is secured; at a uniform air rate, therefore, the exit air temperature is decreased. This effect is observed in the runs at different constant air rates and with the two different plate spacings.

If it is desired to secure a constant exit air temperature, the air rate should be varied accordingly. Thus, if input drops, the air rate should be decreased. This means, of course, that the efficiency would also drop, as would be predicted by comparing efficiency at two different rates in Figure 5 or Figure 6.

Figures 30 and 31 show the basic reason for the close parallelism between gross and net efficiencies observed in many of the preceding figures. It is seen in Figure 30 that within experimental error there is practically no change in gross efficiency even with a change in solar declination of almost 15 degrees. Such a change in declination corresponds roughly to the change in solar position during the period from the middle of October to the middle of February. An entire similar independence of net efficiency on solar position in this range is seen in Figure 31. It is naturally expected that net efficiency would not be affected by change in declination because the calculation of net efficiency involves the deduction from total input of reflection loss, the only loss appreciably affected by solar position. The nearly horizontal lines in Figure 31 bear witness to the validity of this conclusion. The identical trends in the gross efficiencies observed in Figure 30 show that they too are relatively unaffected by seasonal chang in solar position. Comparisons of performance, whether in terms of gross or net efficiencies, must therefore show the same general results.

The lack of strong dependence of gross efficiency on entrance temperature an solar position (season) is fortunate in that efficiencies can be secured in the winter, which are approximately the same as those secured in the summer. Furthermore, although most of the reported runs were made in the fall, the results obtained should apply very satisfactorily to winter operation. If the flow rate is maintained approximately constant, exit air temperatures are of course lower in the winter because of the lower solar input, and if it is desired to hold the exit air temperature constant, the air rate must be reduced in the winter. This latter step would lower the winter efficiency, so a balance of efficiency against exit temperature would have to be established in order to obtain optimum operation.

IX CONCLUSIONS

A. Fundamental

1. The principle of solar heat collection which has been theoretically devised by Miller (10) is workable on a practical scale.

2. Although performance of the full scale collector is not as good as theoretically predicted, solar heat can be recovered in the form of hot air, with an overall efficiency of 35 to 40 per cent and an average daily exit air temperature of approximately 110°F.

3. During periods of high heat input, as in the middle of the day, exit air temperatures of 175° can be secured with an overall efficiency of 35 to 40 per cent.

4. By reducing the air rate to a comparatively low value, air temperatures of at least 225° can be secured, but at an efficiency reduced below 10 per cent.

5. Heat recovery efficiencies which have been secured are comparable with or greater than those previously obtained by the use of other methods.

6. The optimum plate arrangement is with 2/3 overlap, 1/3 of the top surfac coated black, 1/4 inch spacing between plates, and cover plates forming a complete enclosure.

7. The optimum tilt of a collector for solar heat collection at 40 degree latitude, from October through May, is 43 degrees; optimum tilt in other localities is approximately numerically equal to the particular latitude.

8. Only a small reduction in radiation collected is observed when the tilt is greatly reduced; a tilt of 27 degrees at 40 degree latitude permits the collection of nearly 92 per cent as much heat as a unit of the same size would collect at a 43 degree tilt.

9. A collector tilt of 47 degrees is the optimum at 40 degree latitude for winter heating of a house having collector area and heating requirements similar to the experimental house and having a unit storing heat for one day; optimum tilt in other localities would be the latitude plus approximately 7 degrees.

10. As long as the problem of plate breakage remains unsolved, practical application of the equipment cannot be made.

11. The immediate cause of plate breakage is thermal stress resulting from unequal heating of the plates; the manner in which thermal stress initiates breakage is unknown.

B. Secondary

1. Air flow through the unit is not streamline as predicted by Miller, but eddying because of the high temperature difference between plates.

2. The time lag in the outdoor units, because of heat storage in the unit itself, is between one hour and 1.5 hour.

3. In the collector tested, 1 cu. ft. of air supplied per minute for each square foot of collector results in a heat collecting efficiency of about 35 per cent, if the atmosphere is reasonably clear.

4. Any decrease in solar input, such as that caused by clouds, causes no change in efficiency if the air rate is not altered, but results in a lowered exit air temperature.

5. Heat recovery efficiency is only slightly affected by changes in entrance air temperatures.

6. On a clear day, in the test location, solar reflection and heat convection from the cover each constitute about one-third of the losses; radiation from the cover plate and conduction through sides and bottom of the unit make up the balance of the losses.

C. House Unit

1. A solar heat collector of the type herein described can be installed on the roof of a suitable house and employed in conjunction with the standard hot air heating system to supply a portion of the heat required by the house.

2. The household solar heating unit can be regulated entirely automatically so that when heat is required in the house, fuel will be used in the furnace if insufficient solar heat is available.

3. Although the operating variables have not yet been fully regulated to secure the optimum performance, approximately 20 per cent fuel saving has resulted during one winter from the use of a solar heat collector, covering about one-third of the roof, in the one dwelling installation.

4. Results of the operation of a water heater, utilizing the solar heated air, although based on limited data secured over a short period during which heat insulation was not in use, indicate that the domestic hot water needed during the summer can be supplied from the solar unit.

5. The construction employed in the house unit, shown in Figure 3 and Plate 2, is the most practical yet devised and is superior to the construction of the laboratory unit.

6. The weather resistance of the collector employed in the house installation was good; snow slid from the roof, a screen gave protection from hail in the summer, and high wind caused little damage.

D. General Conclusions and Indications

1. Cheap heat storage, although not studied experimentally, can be provided in order to store excess heat collected during the day for use at night.

2. Use of the solar heat collector without the auxilliary heat storage unit is probably impractical in most locations.

3. With proper adjustment and insulation of the experimental house unit, approximately 34 per cent of the annual heating load should be carryable by the solar unit alone; with the installation of a heat storage unit of a size adequate for overnight storage of nearly all the heat collected which is required by the house, the combined collector and storage unit should be able to carry approximately 55 per cent of the annual heating load.

4. A well insulated chamber containing approximately 6 tons of crushed rock, coke, staggered cinder block, hollow tile, or similar material would provide heat storage capacity adequate for the requirement set forth above; the dimensions of such a chamber could be roughly 3 ft. x 3 ft. x 15 ft.

5. As an alternative, heat storage in the form of heated water could be provided.

6. The advantage in providing sufficient storage material to store heat for two or three days is slight and is probably more than offset by the increased cost of the larger storage unit.

7. When heat is not being supplied to the house, the recirculation of air from solar heat collector to the storage unit and back to the collector would be advisable, because this procedure would permit higher storage temperatures and greater heat storage per unit weight of storage material.

8. Heat recovery efficiency should not be appreciably different in widely separated localities, provided that the same air rates are employed and the same area of collector is presented normal to the sun's mean position at the particular latitude.

9. In general, locations in which greater winter sunshine or lower heating loads prevail, (as in most sections further south than Boulder), greater fuel savings will result if collector and storage sizes are made the same as herein described; for the same savings, the units can be smaller.

10. Economic advantages of installations north of the 40th parallel of latitude are doubtful, unless in regions of unusually favorable weather conditions or high fuel costs.

11. The applicability of the solar heated air as the energy source in the operation of an absorption refrigerator type of air conditioner is possible; air cooling should thus be cheaply obtained in the regions where most needed.

12. The cost of a complete solar heating installation under the conditions encountered in the experimental units should, on a large production basis, be no more than $500 over and above the cost of the installed standard hot air furnace, ducts, and auxilliaries.

X STATUS OF SOLAR RESEARCH PROGRAM

At about the time experimental work on the laboratory unit was being completed, an application for a patent covering the construction and operating features of the solar heating apparatus was made. This application is in the name of George O. G. Löf, and was filed as no. 632,504 on December 3, 1945. A copy of the patent application is included in Appendix B of this report. The rights to the patent were then assigned to the University of Colorado on March 5, 1946, and a non-exclusive, royalty-free license was granted to the government on March 18, 1946, under terms of the contract.

Simultaneous with the filing of the Löf patent application, K. W. Miller applied for a basic patent covering the fundamental principles of the solar heating apparatus. The serial number of his application is 632,386. A non-exclusive, royalty-free license to the use of Miller's patent was then granted to the government and the patent rights were subsequently assigned by him to the University of Colorado on March 21, 1946.

Following the formal closing of the WPB contract on November 30, 1945, the University of Colorado continued the work without outside sponsorship. Most of the activities during this period were in calculating the extensive results of previous tests, regulating the operation of the house unit, and studying its performance, organizing and writing this report, preparing the patent application, and making plans for future work.

On March 1, 1946, the Engineering Experiment Station of the University of Colorado commenced a new, two-year research and development program under private sponsorship. Studies to be conducted include the determination of causes and remedies for plate breakage, development of suitable heat storage equipment, improvement in collector design so as to permit large scale production and use of solar heating units, and determination of the most advantageous ways and means to use solar heat in operating air conditioning systems.

XI APPENDIX

XI APPENDIX

A Preliminary Investigations

I. Development of Paint Composition

The paint used on the first set of runs on the outdoor unit consisted of 30 grams of lamp black per 100 cc of solvent containing 85% bronzing fluid and 15% turpentine by volume. This paint did not prove satisfactory because of its low adhesion to the glass.

Seven paints were then mixed as follows:

No.	% by Volume Turpentine	% Bronzing Fluid	Lamp Black g/100 cc
1	90	10	15
2	80	20	15
3	70	30	15
4	60	40	15
5	70	30	10
6	70	30	20
7	70	30	25

The reflectivity of a coating of paint on glass was then measured by a photometer. Each of the above paints was observed to have the same reflectivity hence the same degree of blackness. Each paint coating was then tested for adhesion and it was found that paints 4 and 5 were the most difficult to remove. The composition corresponding to 4 was then chosen for subsequent work.

II. Aluminum Foil Cement

After several attempts to find a good adhesive with which to attach the aluminum foil to the glass plate on the indoor unit, it was found that the best adhesion was secured with white shellac. Foil was therefore cemented to the plates of the outdoor units with this adhesive.

Some peeling of the foil from the plates of the outdoor units has been observed, particularly after prolonged use. However, with a relatively thin foil, this fault is minimized. It is felt that if the foil is finally found desirable or necessary in the finished units, a better adhesive should be developed.

III. Transmissivities of Commercial Glasses

A series of transmissivity tests of several samples of glass furnished by most of the large glass companies was conducted early in this investigation (13) and is summarized below.

The transmissivities were measured by placing the sample between the sun and a pyrheliometer placed in a horizontal position on the roof. The glass was held horizontally and no attempt was made to obtain normal illumination since only a relative transmissivity figure was needed. The solar intensity was reobserved with the pyrheliometer before and after the glass was placed in position, and the intensity of the light passing through the glass was recorded. The percent transmitted was then calculated by the following formula:

$$\% \text{ transmittance} = \frac{\text{intensity passing through} \times 100}{\text{average solar intensity before and after}}$$

The results of the tests are as follows:

Company	% Transmission
A	79.0
B	77.5
C	79.3
D #4	83.8
#1	83.5
#2	83.3

Glass from Company D sample Number 4 was chosen for use in the unit.

IV. Survey of Diffuse Radiation

The total radiation measured by the Eppley pyrheliometer is composed of direct and diffuse radiation. To determine the necessity of considering diffuse radiation in calculating reflection losses from the unit, measurement of the relative magnitudes of the two components of radiation was made by the standard procedure. A metal disk, three inches in diameter, was placed between the sun and the pyrheliometer, three feet from the thermopile. Shading of the thermopile was continued for about three minutes until the electromotive force induced in the thermopile by the diffuse radiation reached a constant value. Removal of the disk allowed measurement of the total radiation.

Measurement of diffuse radiation was made during several clear days. Data for one day appear below:

Date	Solar Time	Total Radiation, mv.	Diffuse Radiation mv.	Per cent of total
9-6-1944	11:05 A.M.	8.7	0.35	4.0
	12:55 P.M.	8.65	0.5	5.8
	2:50 P.M.	6.35	0.5	7.9
	4:00 P.M.	4.1	0.4	9.8

The above results show that diffuse radiation was a small fraction of the total radiation received from a clear sky. It was, therefore, neglected when data were secured during clear hours. Obviously, radiation received during hours when the sky was overcast was entirely diffuse. Reflection losses from the collector during cloudy hours were calculated by assuming that the effective angle of incidence of the diffuse radiation falling on the collector was constant at 58 degrees (9). From Figure 42, the sum of transmittance and absorption of three glass plates for radiation striking the surfaces at an angle of incidence of 58 degrees was found to be 72.5 per cent of the total radiation. This sum of 72.5 per cent was equivalent to a reflectivity of 27.5 per cent. Radiation received from a cloudy sky was therefore considered to be reduced by 27.5 per cent, regardless of solar position.

XI Appendix

B Patent Application

The following patent application was made on December 3, 1945 under the Serial Number 632,504. It is quoted here in full, exclusive of the claims.

"This invention relates to solar heating apparatus and methods and more particularly relates to solar heating systems for household and similar installations.

"Solar heating and the use of solar heat traps has been well known and extensively used for such purposes as greenhouse heating and the like, but in the past, little effort has been made to utilize this heating source effectively in household installations. However, in recent years considerable study of the subject has been undertaken and with the changes and innovations being incorporated i present day architecture, solar heating systems are now recognized as a possible adjunct of future home building.

"The present invention represents the culmination of a series of investigations undertaken to provide a suitable system for household heating and the like, which is adapted both for installation in existing structures and also for incorporation in new construction.

"It is an object of the present invention to provide a simple, efficient and economical method of heating homes or similar structures.

"Another object of the invention is to provide a simple, efficient and economical method of conditioning the circulating air of homes or similar structures both as to temperature and moisture content.

"A further object of the invention is to provide a simple, durable and economical heating system addpted for installation in homes or similar structures, which is adapted to utilize the maximum effect of solar heating as an energy source in the temperature regulation, water heating, and other appliances of the heating and water supply installations of the structure.

"Still another object of the invention is to provide a solar heat trap adaptec for installation in existing homes or the like, which may be utilized as a heat exchange medium to heat air, water or other fluids.

"Other objects reside in novel combinations and arrangements of parts and in novel steps and treatments, all of which will be described in the course of the following description.

"The present invention resides in the discovery that a solar heat trap may be provided in household installations or the like, which contains a plurality of zones or passages, within which heat rays are caused to travel in opposed direction between heat transfer surfaces defining the passages or zones. Within these zones, a fluid, such as air or water, is circulated in contact with the heat transfer surfaces and after heating in this manner, the fluid is circulated to other portions of the structure to be there utilized as a source of heat for the structure

or for appliances, the operation of which is essential in the use of the structure. Preferably, there is also incorporated in the system, means for limiting heat radiation within the confined zones and other means for storing excess heat released through said zones.

"Having thus described in general the features of the present invention, reference will now be made to the accompanying drawings illustrating typical embodiments and practices of the invention. In the drawings, in the several views of which like parts have been designated similarly

Fig. 1 is a vertical section through a structure utilizing the features of the present invention;

Fig. 2 is a vertical section through a heat trap unit embodying features of the present invention;

Fig. 3 is a vertical section through another form of heat trap unit also embodying features of the present invention;

Fig. 4 is a fragmentary section through a heat trap installation illustrating a preferred construction for use in existing structures;

Fig. 5 is a similar section illustrating a preferred arrangement for installation in structures under construction;

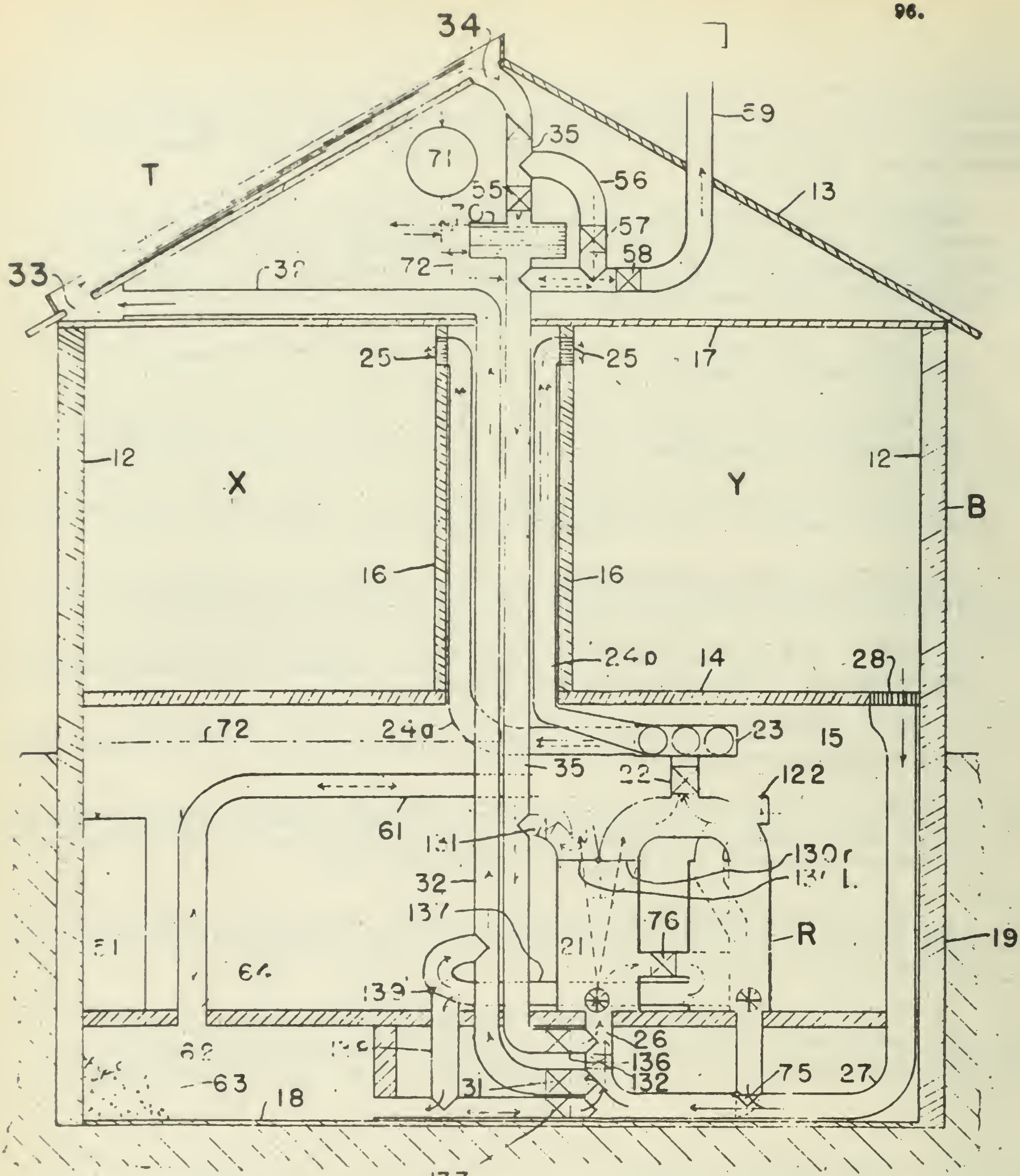
Fig. 6 is a similar section illustrating a preferred installation built as a prefabricated unit;

Fig. 7 is a side elevation of one of the heat exchange elements utilized in the heat traps of the present invention illustrating a preferred method of surface treatment and drawn to an enlarged scale;

Fig. 8 is an enlarged perspective view partially broken to show interior construction in section of one form of heat trap used in the practice of the present invention; and

Fig. 9 is a schematic assembly view showing arrangement of parts in an air conditioning unit adapted to be operated by a solar heat trap of the type shown in Fig. 1 in a circulating system of the type shown in Fig. 1.

"Fig. 1 illustrates an installation in a house or other habitation in which the various forms of solar heat utilization of the present invention have been combined in a single installation. It will be understood that for most purposes, liquid and air room heating installations will not be combined within the same structure, although under certain circumstances, it may be necessary or advisable to do so, for which reason both forms have been described or illustrated in the drawings. The air conditioning circuit illustrated is intended for use independently of or as an adjunct of the household heating system and a typical installation has been shown in Fig. 1. Also, the use of the solar heat source for heating the domestic water supply has been illustrated and this may either be a



LEGEND

Internal		External	
Hot air	----→	Hot water	—·—→
Normal air	—·—→	Cold water	——→
Chilled air	—··—→		

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FIG. 1

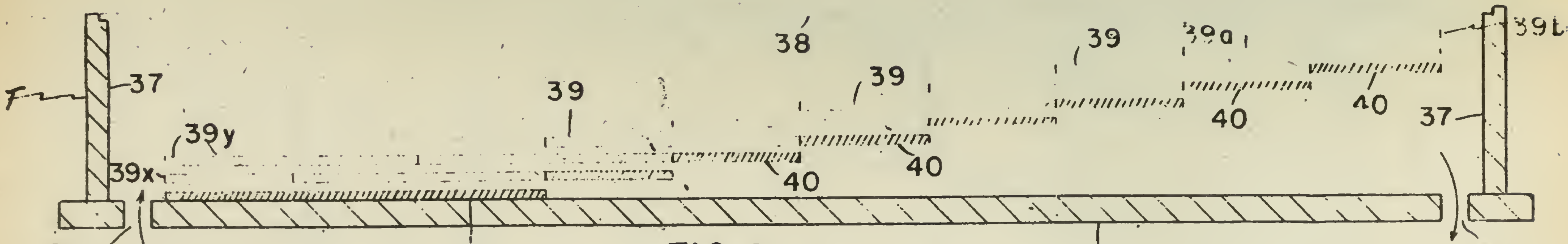

FIG. 2

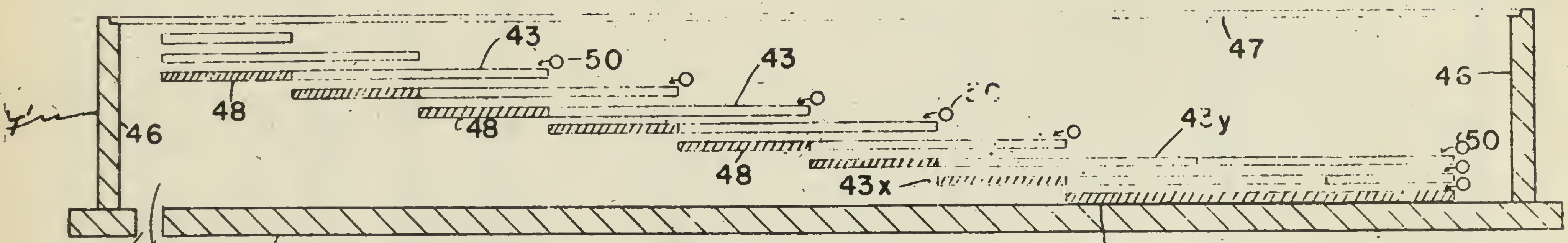

FIG. 3

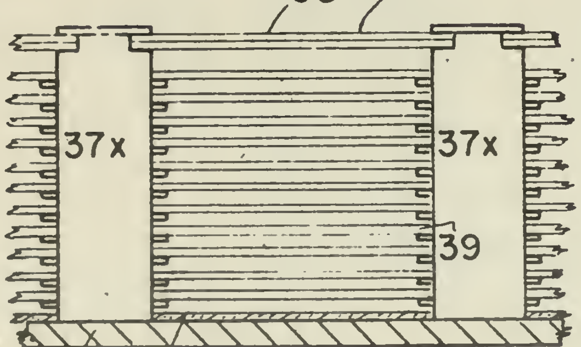

FIG. 4

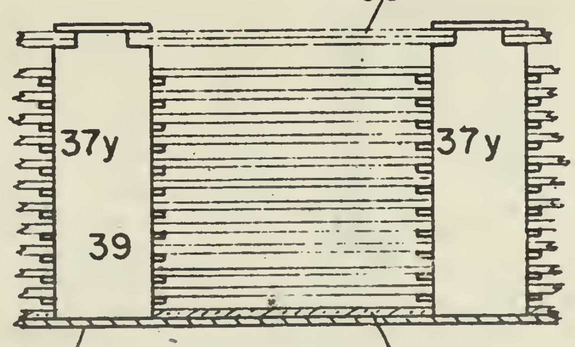

FIG. 5

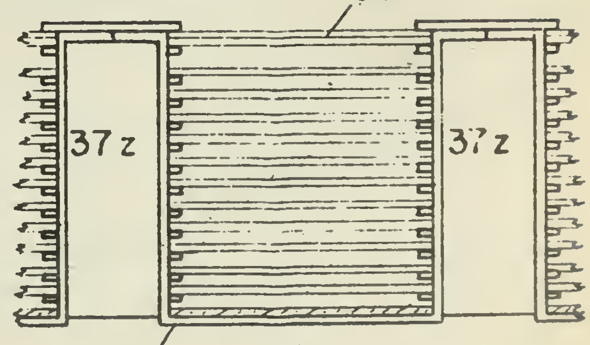

FIG. 6

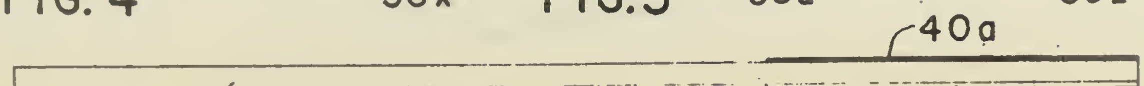

FIG. 7

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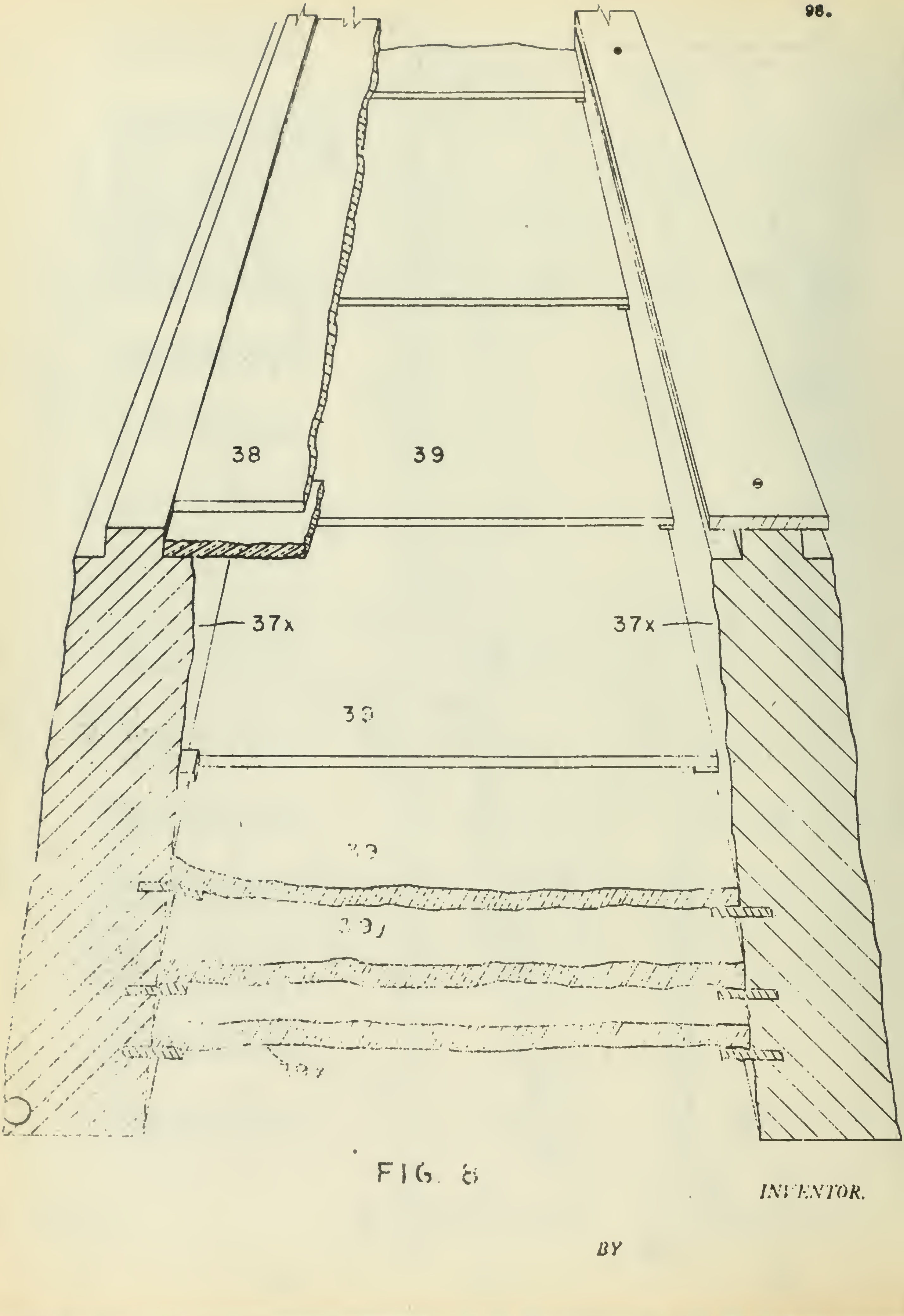

FIG. 8

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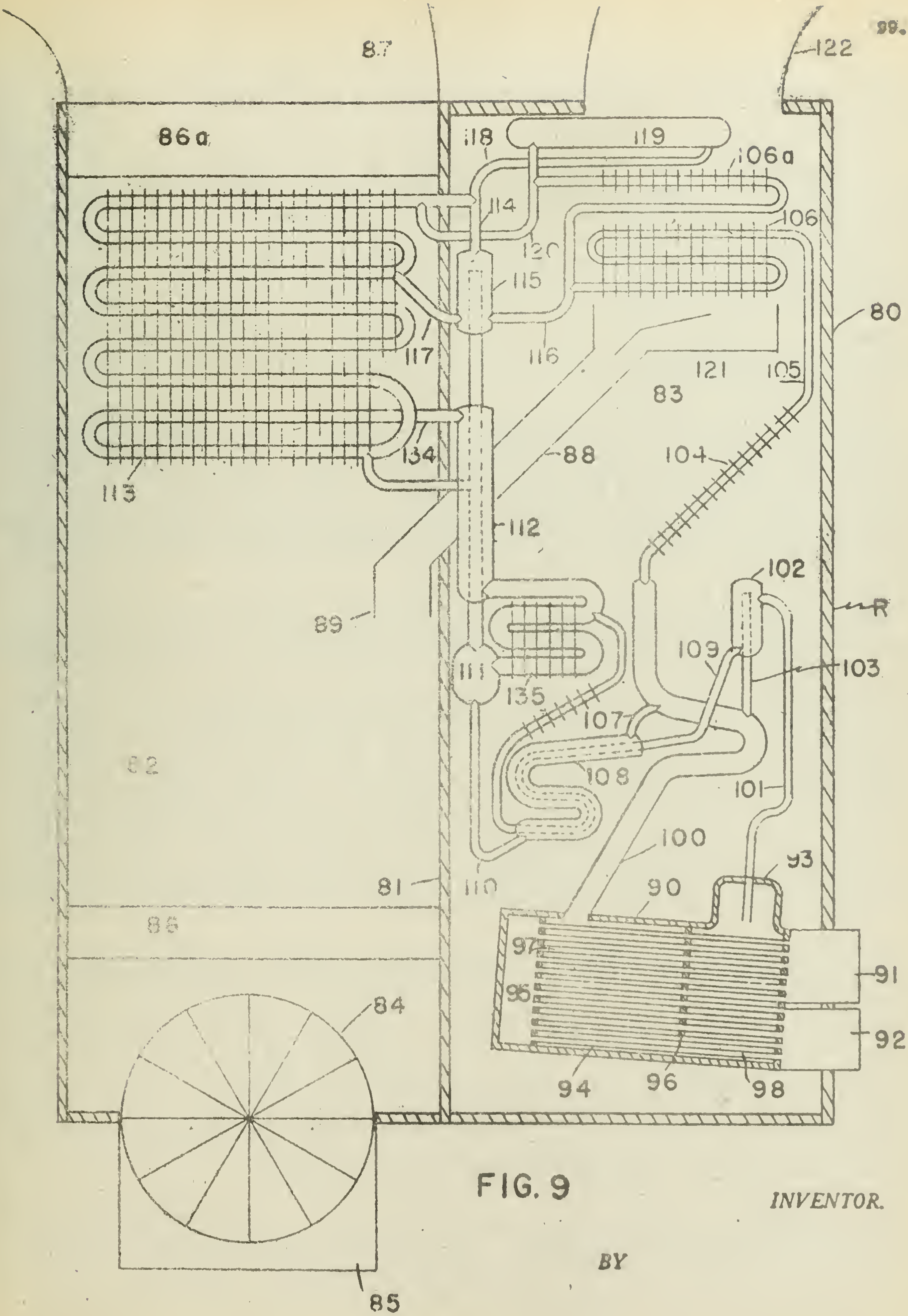

FIG. 9

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separate installation or incorporated as an adjunct to the existing household installation.

"In Fig. 1, a building designated generally by the reference letter B is intended to illustrate a home or business structure and as shown embodies a two story construction, but which may be any number of stories within the capacity limits of the system to heat. As illustrated, the building has upright walls 12 covered by a roof 13 with a floor 14 partitioning the interior space into a basement 15 and other partitions 16 cooperating with the floor 14 and a ceiling 17 to divide the first floor space into a plurality of rooms designated X and Y. The basement is provided with a floor space 18 of any suitable material, such as concrete, and the foundation structure 19 for the building also may be any suitable material such as concrete.

"A section of the roof 13 preferably located so as to have good exposure to the sun in both summer and winter is suitably apertured to receive a heat trap T, the details of which will be described hereinafter. When the heat trap T is to be utilized as a heating source for the building, it may be necessary to have it operating in conjunction with a stand-by heating plant and utilize a common system of conduits, outlets and the like and systems of this type have been illustrated in Fig. 1.

"Within the basement 15 or in some other suitable area of building B, a furnace 21, preferably a blower-type circulating air heater, is located to deliver heated air through a valve controlled conduit 22 into a distributor 23 from which a series of conduits 24a and 24b deliver the heated air to room outlets 25. The furnace 21 also has a valve controlled intake 26 preferably supplied by a cold air duct 27 having its inlet opening 28 located as in the floor of room Y to provide a closed circuit air circulating system. However, it should be understood that if desired, an open circuit circulation may be employed, in which case the duct 27 would have its inlet located to receive atmospheric air from a point outside the building B. The furnace 21 may utilize any suitable type of fuel, such as coal, oil or gas and at such times, for example during the night hours, when the system may not utilize solar heating, the furnace is operated as in conventional installations to heat the rooms X and Y through discharge of heated air through outlets 25.

"In utilizing the solar heating source in this circuit, at least a portion of the cold air entering through duct 27 is by-passed from the intake 26 through regulation of suitable valves 31 and 132 into a conduit 32 which in turn connects with the lower portion of heat trap T through a suitable opening 33 preferably located at the lower end of said trap. At the upper or opposite end of said trap, another opening 34 connects with a valve controlled conduit 35 which delivers heated air from the trap into furnace intake 26 and thence through the circulatory system as hereinbefore described. With this understanding of the general arrangement of the circulatory system, reference will now be made to the details of construction of the heat trap and its function and operation.

"The arrangement shown in Fig. 2 illustrates a preferred construction for installations of the type just described. The unit comprises a framework having a bottom 36 and side walls 37 preferably forming a rectangular enclosure. The bottom and side walls may be formed of any suitable material, such as wood or metal, and preferably are of a dimension not requiring too great an unsupported

surface for the glass parts now to be described. The top closure for the heat trap comprises a glass plate 38 supported at its sides and ends by the side walls 37. Although not shown, it will be understood that where it is necessary to protect the cover plate 38, as from hailstones or the like, a suitable wire screen or other protecting means permitting light penetration may be mounted in overhanging relation to plate 38. Within the enclosure thus formed, a series of glass plates 39 are mounted in spaced and substantially parallel relation and preferably staggered lengthwise of the enclosure in the manner shown. Each of said plates except special end plates 39a and 39b are provided at one end with an opaque and essentially nonreflecting area 40 preferably formed as by covering the said surface of the glass with black paint.

"The opaque area thus provided preferably will be of a uniform length which is an even division of the total length of the plates 39. As here shown, the length of the opaque area is one third the length of the entire plate except at the lower end where one plate 39x is four times the length of its opaque area and another plate 39y is five times the length of its opaque area. Also at the lower end of trap T, the surface 41 of bottom 36 underlying the transparent portions of plates 39x and 39y is blackened in a manner similar to the plates 39, or a sheet of material, mounted on the bottom will have its upper surface blackened, as shown. Through the arrangement just described, solar heat rays entering through plate 38 are at all times directed against a blackened surface throughout the extent of the plate assembly. Obviously, the length of the opaque areas may be increased or decreased for a given sized plate and as long as the overall effect is similar to that just described, satisfactory results will be obtained.

"While the arrangement of the heat trap T thus far described is adequate for most purposes, more satisfactory results will be obtained by providing a reflective medium throughout the opaque area of the glass plates 39. Such an arrangement has been shown in Fig. 7, in which a glass plate 39M having an opaque area provided by applying black paint 40a to its top surface has the portion of its undersurface underlying paint 40a covered with a reflective material such as aluminum foil 41. Through the use of the light reflective medium, a uniform reflective surface or surface effect is obtained which serves to retard heat losse occasioned by temperature differentials between glass plates 39 and bottom 36.

"The arrangement of opaque and transparent surfaces just described has the further advantage that as viewed from underneath, the blackened areas provide a uniform reflecting surface. Because of this, the heat emanating from the blackene surfaces by reason of the stoppage of the solar heat rays travels in a reverse direction to said rays with the result that with respect to any given plate, both its top and bottom surfaces are substantially heated. Therefore, when cold air flowing through duct 32 enters into the enclosure and passes between the several plates 39, a heat exchange action results which imparts an upward flow to the entering air and aids its travel to a point of escape at opening 34, although the primary air movement is induced by the blower in furnace 21. Due to the aforesaid heat exchange action, a substantial increase in temperature results so that the heated air passing through opening 34 comprises an adequate heating medium for the building. This air travels through conduit 35, enters furnace intake 26 and is then moved by the blower actuation to the distributor 23 and circulated through the rooms in the manner hereinbefore described.

"With either type of heat trap as herein described, the parallel arrangement of the glass plates having opaque and transparent areas provides a multiple air film thermal insulation, as well as providing heat transfer surfaces to the air flow between the plates heated by solar radiation. Consequently, the solar heat is transformed to warm air at temperatures high enough to be adequate for house heating purposes even in extremely cold weather, and by providing sufficient capacity through the use of a battery of such heat traps, preferably arranged in adjoining relation, it is possible to generate sufficient heat units to function as a heat source for various household appliances, in addition to supplying the required amount of heat for house heating purposes.

"For most purposes, the heat trap will be utilized to heat a circulating air stream in the manner just described. However, it is practical to utilize this heat source in heating other fluids, such as water for example. Fig. 3 illustrates such an arrangement in which the trap T' is arranged to have liquid flow between the parallel plates 43 which are generally similar to the plates 39 of Fig. 2 and pass from the enclosure through a lower outlet 44 in the bottom 45 of an enclosure similar to trap T, having side walls 46 and a transparent cover plate 47. As in the other form, the plates have opaque areas 48 preferably applied by black paint and the surface of the bottom 45 underlying the transparent portion of end plates 43x and 43y is covered with black paint as shown at 49.

"In this form of assembly, the upright walls 46 and bottom 45 will have to be joined in water-tight relation to confine the liquid and prevent leakage into other parts of the structure in which the heat trap is located. As installed, trap T' will be located in an inclined position with the outlet 44 constituting the low point through which the released liquid will flow. The liquid is introduced into trap T' by a series of pipes 50 supplied from one or more headers (not shown) and the conduits have a series of jet or spray outlets from which liquid passes on to the upper ends of the respective plates 43, 43x, 43y and 49.

"These plates are heated in the same manner as previously described through the arrangement of transparent and opaque surfaces and the liquid flowing downwardly along the plates is subjected to an intense heat-transfer action with the result that when it collects at the lower end of trap T' and passes through outlet 44, it is at a temperature adequate for the requirements of the household system. Any vapors generated in the heat exchange action will rise in contact with the undersurface of the overhanging plate and thus are subjected to further heat exchange action. Other vapors passing out of the upper ends of the passages between parallel plates and heated air within the enclosure tend to collect in the upper end of the enclosure and act as an additional heat source to assist in the overall heat transfer action of the unit. Moisture condensing at any point in the upper part of the enclosure will ultimately fall on to one of the heat transfer surfaces and then descend along same to reach the outlet 44.

"In use, the outlet 44 will be connected with a suitable conduit to deliver the collected contents to a storage receptacle or some other point of ultimate use within the structure. Where the units of this type are to provide the circulating water for a household radiation system, for example, it usually will be necessary to have several of such units arranged as a battery to provide the necessary capacity. When a lesser quantity of heated water is required, as for example in supplying a hot water storage tank, a single unit will provide the required amount of heated water.

"From the foregoing description, it will be apparent that the structural arrangements of the present invention may be utilized in heating a variety of fluids of which air and water are typical. Referring again to the arrangement of the circulating system shown in Fig. 1 hereinbefore described, it will be apparent that the various controls of valves, dampers, blower and the like may be automatically controlled as by thermostat regulation, for example.

"It will be apparent that in the operation of the system thus far described, the heat trap T on clear days may produce an excess amount of heated air during at least a portion of its operating period. Two satisfactory methods of handling this excess have been shown in Fig. 1. For example, in the roof installation, a valve 55 may be operated to close conduit 35 and cause the heated air passing therethrough to enter a by-pass conduit 56. Through suitable regulation of other valves 57 and 58, a portion of the hot air flowing through conduit 56 will pass into a stack outlet 59 while the remainder will flow back into conduit 35 and thence pass to furnace intake 26. If the aforesaid by-pass arrangement is not being utilized in the system, valves 57 and 58 are closed and valve 55 is open to permit the direct flow of air through conduit 35. After delivery into furnace 21, a portion of the heated air may be diverted through suitable damper regulation and passed into a conduit 61 which discharges into a heat storage bin 62.

"Preferably, this bin is sealed and insulated from the atmosphere except for the inlet and outlet openings hereinafter to be described and a large portion of the volume of the bin is filled with a heat absorbing and retaining material 63 which preferably is a loose or spaced solid, such as sand, gravel, or stacked brick, but which may be a fluid, such as tar, oil, water or the like. Consequentl when the excess of heated air discharged by furnace 21 into conduit 61 is delivere through a suitable opening 64 into bin 62, the bed 63 is gradually heated and functions as a heat storage unit of the system, and the air, after heat extraction, is recirculated to the heat trap through conduit 32. Subsequently, when heated air is no longer supplied to the furnace from heat trap T, warm air flows back through conduit 61 until diverted by a gate 130L and through a branch pipe 131 to pass into furnace 21 and is then distributed through the heat outlets in the manner previously described. The air after passing through rooms such as X and Y then returns to bin 62 via duct 27 with suitable adjustment of valves 75, 132, 31 and 133. Thus, it will be seen that excess heat produced in the operation of the solar heat trap may be utilized in the household system, if desired, or if not, may be wasted to atmosphere to prevent undue heating of building B.

"It will also be desirable to provide an arrangement for heating a portion of the domestic water supply of the building at such times as the heat trap is in operation. As shown in Fig. 1, this is accomplished by providing a heat exchange unit 70 mounted about conduit 35 with a portion of the heated air passing therethrough diverted through a system of flues or similar water jacketed passages to heat the contained water of the unit. A portion of the heated water then passes to a storage tank 71, which tank has a cold water return to unit 70. The remainder of the heated water passes through a line 72 which empties into storage heater 51 located in the basement of building B. Suitable draw-off connections may be provided for both tank 71 and storage heater 51 and it will be understood that whenever the heat trap is unable to supply water at the required temperature, the storage unit 51 will be operated in the usual manner to heat the water required in he household supply.

"The heat trap T also may be utilized to provide cooled, conditioned air when required for distribution throughout the building. To accomplish this, the furnace operation is stopped and the heated air delivered through conduit 35 is passed from furnace 21 into a refrigerator unit R through the opening of a suitable valve 76. After passing through generator 90 of refrigerator R, the air passes through duct 137 to duct 32 and recirculates to trap T, damper 139 being suitably adjusted for this operation. Air to be circulated by refrigerator R is drawn into same from duct 27 by a suitable blower unit, the course of flow of air through duct 27 having been changed by operation of a suitable valve 75. The air stream passing into refrigerator R after being suitably cooled is delivered to distributor 23 and then circulated through the various conduits 24a and 24b to the room outlets 25 to reduce the room temperature of rooms X and Y.

"It will be understood that any suitable refrigerator unit may be used for this purpose and in order to clearly describe the practice of the invention, a suitable air conditioning unit has been illustrated in Fig. 9. This unit comprise a casing 80 having a partition 81 dividing its interior into a cooling chamber 82 and a second compartment 83 in which most of the operating parts are located. The cooling chamber 82 contains a blower 84 preferably having its intake 85 below the casing to receive the returned air flow diverted from duct 27 as previously described. Preferably, compartment 82 will have means for filtering incoming air which may be located at any suitable place, such as the area 86, and a means for humidifying air which may be located at any suitable place, such as the area 86a. An outlet for the air delivered to chamber 82 is provided at its top, as shown at 87, and in the installation shown in Fig. 1, this outlet will deliver the cold air past valve 22 to distributor 23. A conduit 88 having its intake 89 located within compartment 82 extends through partition 81 and discharges its contents within chamber 83 in a manner that will be hereinafter described. The mechanism of this air conditioning unit is of a conventional design of the type used in certain commercial refrigerators except for the generator unit 90 shown in chamber 83, the construction details and operation of which will now be described.

"The generator has an air inlet 91 and an air outlet 92 at one of its ends and adjacent thereto a dome 93. The interior of the generator contains a tube section 94 of the general arrangement of conventional boiler construction. The end of the generator opposite inlet 91 contains a space 95 beyond the end of tube section 94 through which the hot air flows to pass into the return passages of tube section 94. A wall 96 divides the tube section into a main heating portion 97 and a secondary heating portion 98, and assists in forming a pressure head in dome 93.

"This generator unit is used to vaporize ammonia from an aqueous ammonia solution by a heat transfer action and utilizes hot air supplied from heat trap T in the system shown in Fig. 1. To accomplish this, a conduit 100 delivers aqueous ammonia solution into the enclosure of tube section 94 where it is boiled by the heat transfer action and the resulting vapors rise in the main heating section 97 to pass from the unit through the same conduit 100 through which the solution is delivered to the unit. This counter-current circulation has the further advantage of preheating the solution flowing to generator 90. The gases rising through conduit 100 pass into a rectifier 104 which releases freed ammonia into a line 105 supplying a condenser 106 while the water condensed by rectifier 104 returns to conduit 100.

"The gaseous ammonia entering condenser 106 is condensed to the liquid form and passes to evaporator 113 through conduits 116 and 117. Any uncondensed ammonia rises to condenser 106a and the liquid ammonia formed therein passes to the evaporator 113 through conduit 120. The downflowing streams of liquid ammonia meet an upflowing stream of hydrogen gas entering evaporator 113 through conduit 134 hereinafter described, the liquid ammonia being evaporated by said hydrogen accompanied by an extraction of heat from the circulating air in body 82 and thus providing cool air for rooms X and Y. The mixed hydrogen and ammonia vapors then pass from the evaporator through conduit 114, and the inner conduits of 115, and heat exchanger 112, to bulb 111. The gases then pass upward through absorber 135 countercurrent to a downward stream of water hereinafter to be described. The absorber 135 absorbs ammonia from the gases and the remaining hydrogen passes up the outer conduit of heat exchanger 112 to reenter evaporator 113 through conduit 134. The hydrogen reserve tank 119 serves to keep the pressure in the hydrogen system constant during room temperature changes.

"When most of the ammonia has been boiled from the aqueous solution in heating section 97 the solution has a higher density and flows to the bottom of the generator 90 and underneath the baffle 96 to the secondary heating section 98 where further evolution of ammonia occurs. The ammonia gas rises into dome 93 creating a pressure which forces the dilute ammonia solution, hereinafter called water, up through conduit 101 into head tank 102. The ammonia vapor accompanying the water continues down conduit 103 to join the main ammonia stream going upward to the condenser from conduit 100. From head tank 102 the water flows by gravity down conduit 109, through the inner conduit of heat exchanger 108, and into the top of absorber 135. The water then flows down the absorber 135 countercurrent to the stream of mixed gases flowing upward and absorbs the spent ammonia gases hereinbefore mentioned. The aqueous ammonia solution then flows into bulb 111, down conduit 110, through the annular space in heat exchanger 108, and into conduit 100 through the connecting conduit 107. The aqueous ammonia then flows into the generator 90 to complete the cycle. The flow of aqueous ammonia from the absorber 135 to the generator 90 is actuated by gravity from head tank 102.

"The air diverted from chamber 82 which passes through conduit 88 reaches a distributor 121 which directs it across the surfaces of condensers 106 and 106a after which it passes into an outlet 122 to be exposed to atmosphere in any suitable manner as through a stack (not shown).

"In the operation of an installation such as that shown in Fig. 1, it frequently will be desirable to store heat generated at heat trap T without circulating hot air through the rooms of the building, as for example, when the temperature of the room is sufficiently high through a preceding heating operation and additional heat is passing from the discharge of the heat trap. Under these circumstances, a damper 130r in furnace 21 is moved to close the passage through valve 22 to distributor 23 and a damper 130L is opened to allow the gases entering the furnace through intake 26 to pass into conduit 61, the valve in branch conduit 131 being closed. At the same time, damper 75 is moved to shut off the flow of cool air through duct 27 and another valve 132 is closed to block the passage between duct 27 and intake 26 while a valve 133 is open to permit an outward flow of heated air from storage bin 62.

"When so arranged, the heated air leaving trap T flows downwardly through conduit 35 and into furnace 21 through intake 26. Having no escape except through conduit 61 hot air passes through duct 61 into a storage bin 62. Through the opening of valve 133, a pronounced flow of air through bin 62 is obtained with the air returning therefrom entering conduit 32 past valve 31 which also has been open and thus the air returns to the entrance 33 of heat trap T.

"So long as this operation is allowed to continue, the circulating air will be progressively heated, thus raising the temperature of the bed 63 in bin 62 and as this circuit is insulated from other portions and particularly the occupied portions of the building, no appreciable temperature rise occurs in these occupied portions. This circulation will be allowed to continue so long as the solar heat trap is functioning and whenever there is a further demand for heat, either from furnace 21 or air conditioner R or the water heating stages previously described, the circulation can be discontinued and the generated heat made available where required.

"When stored heat is to be used for the operation of air conditioning unit R, hot air is drawn through duct 61 from storage bin 62. This hot air proceeds to duct 35 through duct 131 by suitable adjustment of dampers 131 and 130L. The hot air then flows into furnace 21 and into refrigerator R by suitable adjustment of damper 76 and after being used flows through duct 137 and 138 and back into heat storage unit 62 by suitable adjustment of vane 139 and damper 133. It is possible, if desired, to operate the refrigerator R by applying heat from natural or artificial gas or other similar fuel when the heat storage unit is cold and trap T is cold.

"In the operation of the system previously described, the location of by-pass conduit 56 provides a convenient arrangement to prevent overheating of the storage water supply which otherwise might occur if heat exchange unit 70 were operated at all times when hot air was flowing through conduit 35. Whenever the temperature of the water in tank 71 reaches an established maximum, the valve 55 may be closed and valve 57 opened to permit the flow of the hot air through branch 56 and thence back to conduit 35 without heating the water in the heat exchange unit 70. Preferably, a thermostat control will be utilized to provide automatic regulation at this stage, although manual or other types of operation may be used, if desired.

"In Fig. 1, no attempt has been made to show the insulation of the heat trap, conduit, storage bin and the like. However, it will be understood from the foregoing description that suitable heat insulation may be provided for all of the conductive parts of the system and the insulation to be used for this purpose may be any one of a variety of materials available on the market for such purposes.

"Next referring to Fig. 2, it will be understood that if desired this form of construction may be utilized as a water heating unit rather than as air heating unit as described. In order to do so, it will be necessary to make the entire closure, inclusive of bottom 36, side walls 37 and cover plate 38, into a water-tight assembly and then pump water in under pressure through the opening 33 to effect its movement across the heat exchange surfaces and its ultimate discharge at outlet 34. As the production of such a unit would involve construction difficulties in providing sufficient structural strength to carry the load and to withstand the pressures required in the pumping action, I prefer to use the form of

construction illustrated in Fig. 3 whenever the heat trap is to be used in heating water rather than a gaseous fluid.

"In assembling the heat trap in the roof of a structure, various arrangements may be employed. Where the installation is to be made in existing structure, the arrangement shown in Fig. 4 is particularly suitable and comprises upright walls 37x which cooperate with end walls 37 (not shown) preferably of similar width and thickness to form a box-like enclosure. The enclosure has a bottom 36 of the type hereinbefore described which preferably comprises the roofing material and the interior space of the enclosure is insulated from the building enclosure by a strip or bed of suitable heat insulating composition 36a.

"As previously explained, it is necessary to have some suitable transparent cover for the enclosure and this is most conveniently effected by arranging a plurality of glass plates in tiers or layers with a portion of an end surface of one plate overlying another end portion of a second similar plate, a greater portion of which projects beyond the first said plate in a direction lengthwise of the enclosure. When it is necessary to brace the respective plates to support the weight of the cover assembly, suitable straps or bars of wood or metal may be arranged to bridge the space between the members 37x in supporting relation to the cover plate glass.

"The arrangement just described will be best understood by reference to Fig. 8 which is a perspective view partially broken away to illustrate an assembly of this character. While the arrangement shown in Fig. 8 is illustrative in general of all the arrangements shown in Figs. 4, 5 and 6, it is more exactly a representation of the construction shown in Fig. 4 and consequently has been given corresponding reference numerals. In the preceding description, the arrangement of transparent and opaque surfaces has been described as being embodied in a plate of glass. While this is a preferred arrangement because of the simplicity of construction and assembly, it will be understood that the opaque black areas may be any suitable material which is non-reflecting, and if desired, may be separate pieces arranged in end to end relation with the transparent plates of glass or other suitable composition.

"Where the heat trap units are to be installed in new constructions, it is possible to use the rafters 37y of the roof structure as a part of the trap enclosure, mounting thereon a cover plate 38 in all respects the same as the cover plate 38 shown in Fig. 4 and providing insulation 36a of the type previously described with a bottom portion 36x preferably attached to and in underhanging relation to the rafters 37y. Within the enclosure, the arrangement of transparent and opaque surfaces previously described will be provided.

"Still another arrangement has been illustrated in Fig. 6 in which the enclosure is formed to seat upon rafters 37z and has end walls (not shown) similar to those previously described. A bottom piece 36z encloses the space between two adjoining rafters 37z while the usual type of transparent cover plate will be provided at the top of the enclosure. As clearly shown in Fig. 6, the material of the bottom portion 36z is U shaped in section and fits against the top and side surfaces of the rafter 37z. Through this arrangement, it is possible to order any required number of such trap units and mount same on rafters or other upstanding supports with the respective units joined in side by side relation as indicated in Fig. 6.

"From the foregoing description, it will be apparent that the solar heating system of the present invention is well suited for incorporation in new constructions or in existing structures, and only a minor amount of the habitable space of the house or other structure is occupied by component parts of the system. When desired, the solar heat trap may be supplemented as a heat source by standard type furnaces, water heaters and refrigerators, although with proper capacity in the heat trap and heat storage units such standby appliances will be unnecessary in many installations.

"The accompanying drawings illustrate typical installations for carrying out the purposes of the present invention, and it will be understood that variations in the construction and operation are within contemplation of the invention. Therefore, the construction and arrangement of parts shown and described is not intended to limit the invention

XI APPENDIX

C METHODS OF CALCULATION

I. Calculation of Charts and Nomograph

A. Thermocouple Conversion Chart

Temperatures of couples were recorded by a Brown potentiometric pyrometer which was calibrated and compensated for iron-constantan couples with a cold junction temperature at the instrument temperature. A calibration for iron-constantan couples with a 0°F. reference temperature was used and the potentiometer compensated to allow for the difference between the instrument temperature and 0°F. Chromel-constantan couples, used for measuring surface temperatures, with a cold junction ice bath, were also connected through the pyrometer.

Assuming an average instrument temperature to be 100°F. and constant, a conversion chart was made to give chromel-constantan temperatures from the temperature difference between indicated temperature and instrument temperature. (Figure 32 , Appendix)

For a temperature difference of 90°F.

Indicated temperature = 100°F. + 90°F. = 190°F.

From iron-constantan calibration:

$$t = 190^\circ;\ mv = 5.54$$
$$t = 100^\circ;\ mv = \underline{2.88}$$

$$\Delta t = 90^\circ;\ mv = 2.66$$

From chromel constantan calibration:

At mv = 2.66; t = 108.5°F.

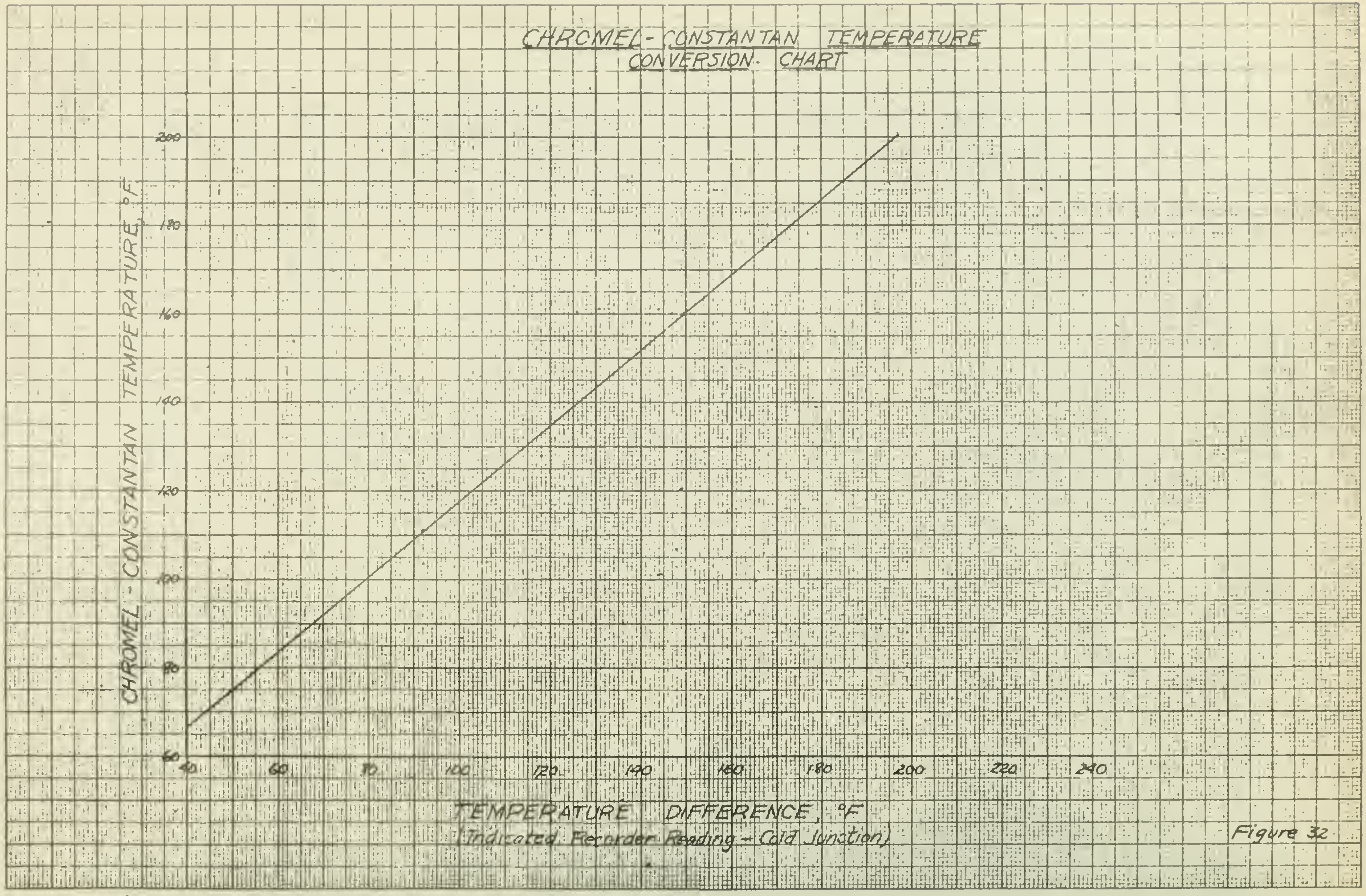
Therefore, 108.5°F. is plotted versus 90°F.

B. Charts and nomograph for calculation of air rates.

1. Rotameter

a. Calibration curves.

Calibration curves for both rotameters were furnished by the Fischer and Porter Company of Hatboro, Pennsylvania. (Figure 33 , Appendix)


CHROMEL - CONSTANTAN TEMPERATURE
CONVERSION CHART
CHROMEL - CONSTANTAN TEMPERATURE, °F
200
180
160
140
120
100
60
40
60
100
120
140
160
180
200
220
240
TEMPERATURE DIFFERENCE, °F
(Indicated Recorder Reading - Cold Junction)
Figure 32

b. Pressure correction

Since the above calibration curve was made to read volume of air at (760 mm. - 70°F.) measured at (630 mm. - 70°F.), correction must be made for measurements at pressures other than 630 mm. Hg. and temperatures other than 70°F: (16)

$$\text{Correction factor} = \left(\frac{P\ \text{actual}}{630}\right)^{\frac{1}{2}} = \left(\frac{610}{630}\right)^{\frac{1}{2}} = 0.984$$

A plot of the factors versus P actual was made. (Figure 34 Appendix)

c. Temperature correction

Similarly, a correction must be applied if the air rate is not measured at 70°F.

$$\text{Correction factor} = \left(\frac{530°R}{460+t'_{actual}\ °F}\right)^{\frac{1}{2}} = \left(\frac{530}{460+100}\right)^{\frac{1}{2}} = 0.971$$

A plot of these factors was also furnished by the Fischer & Porter Company. (Figure 35, Appendix)

2. Nomograph for 3.5" air orifice.

A nomograph for finding air rate from the orifice data was made from the following equation. (Figure 37, Appendix)

$$V = \frac{142.8\ M^{0.515}\ (B-1.87P'-0.935M)^{0.485}}{T^{0.485}}$$

where

M = pressure drop across orifice, inches H_2O
B = barometric pressure, mm Hg.
P' = upstream static pressure, inches H_2O
T = temperature of fluid, °F.
V = volume of air, c.f.m. (760mm-70°F.)

The above formula was derived from the basic orifice equation and a calibration curve for the orifice which was obtained by calibrating the orifice with the above mentioned Fischer & Porter Rotameter. (Figure 36, Appendix)

References (20), (21)

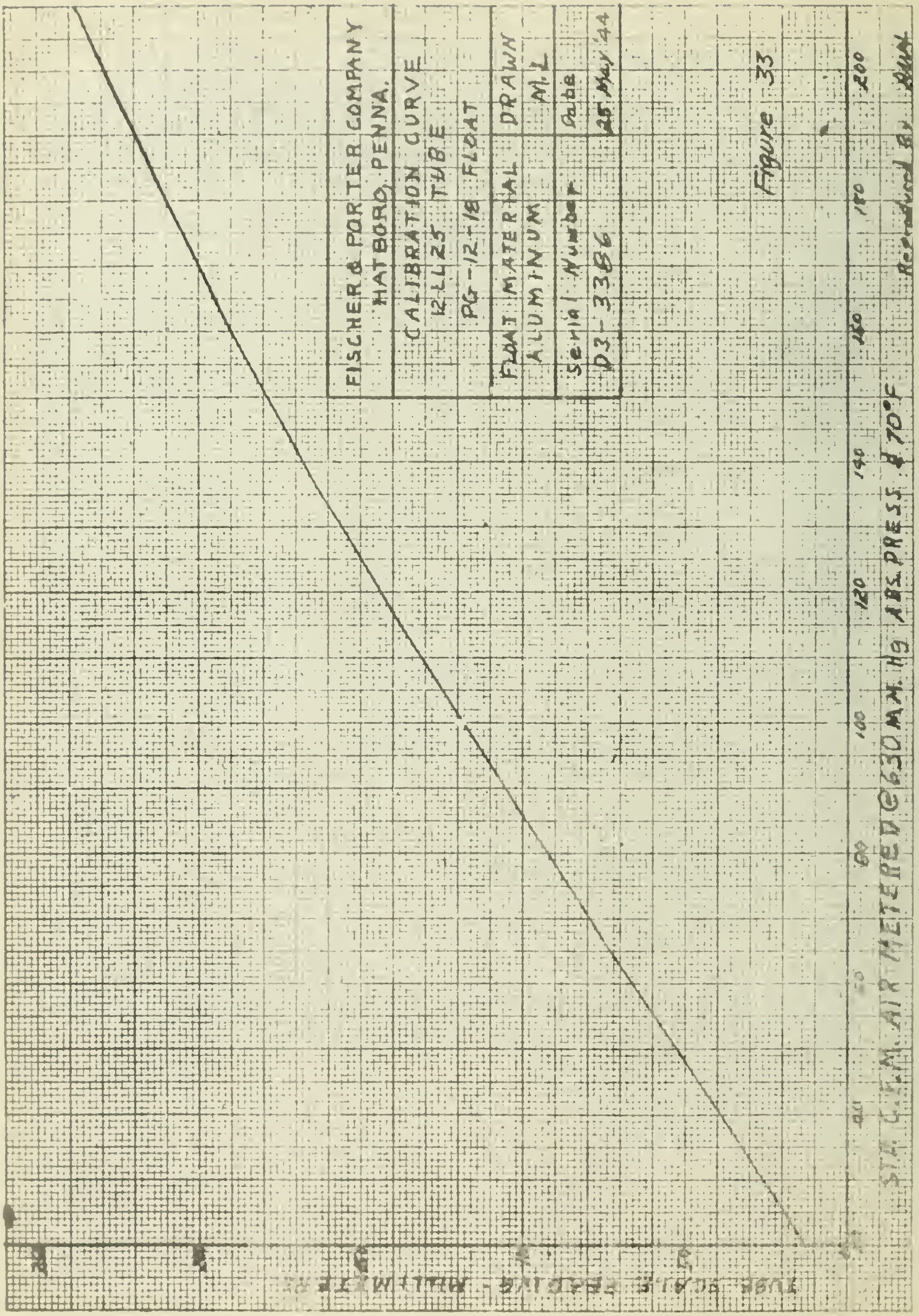
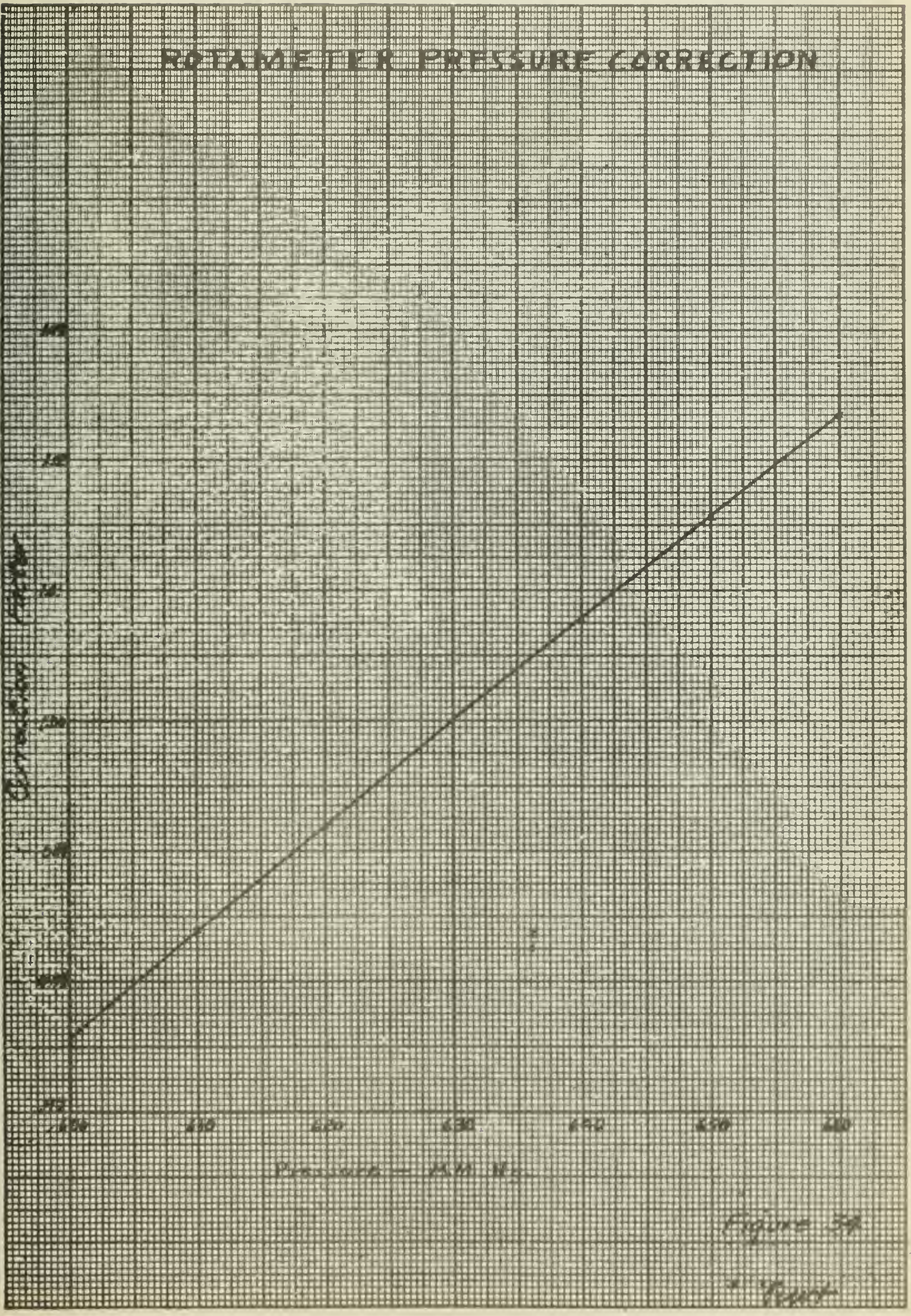
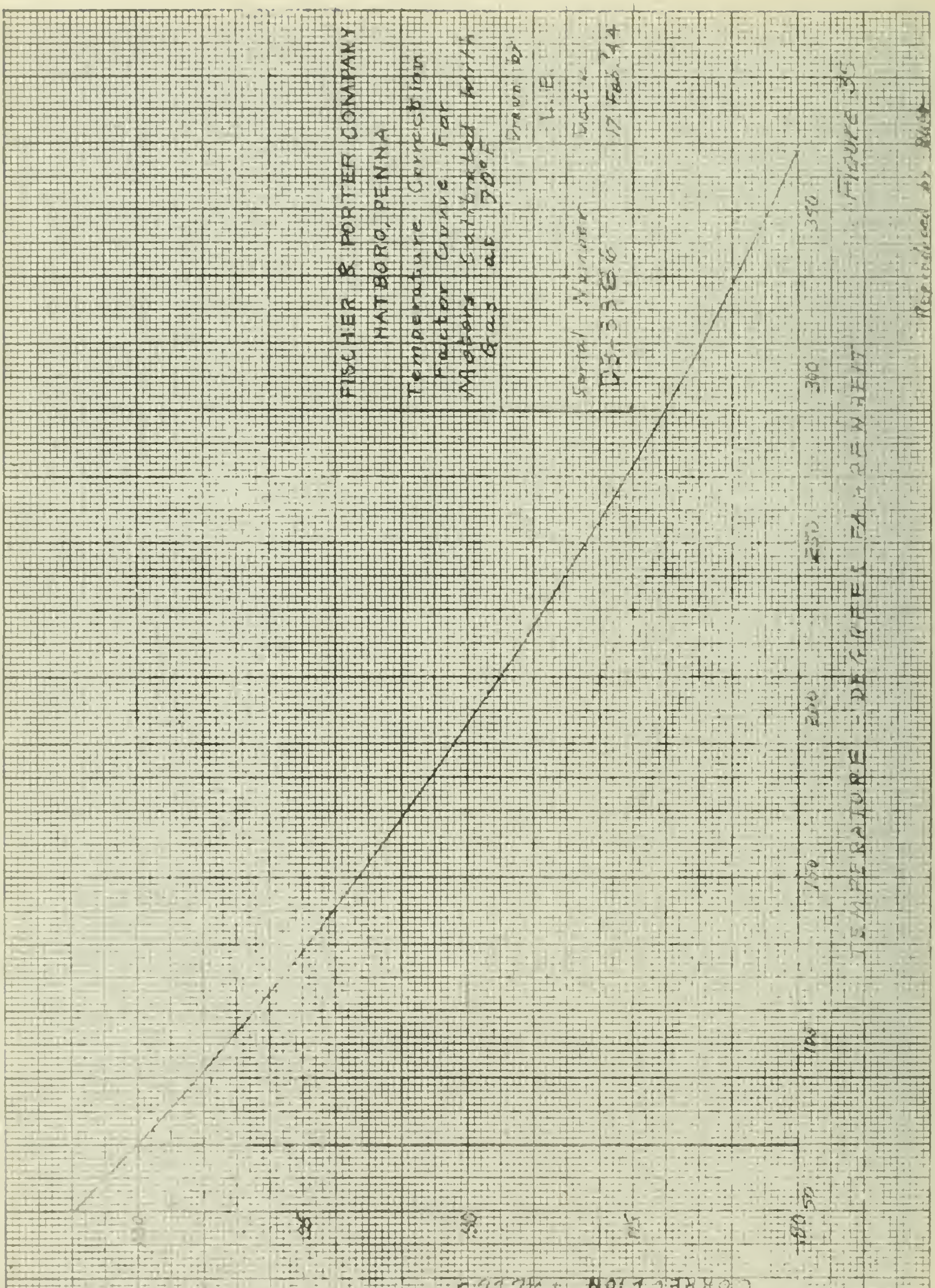

FISCHER & PORTER COMPANY
HATBORO, PENNA.
CALIBRATION CURVE
12-LL25 TUBE
PG-12-18 FLOAT
FLOAT MATERIAL
ALUMINUM
DRAWN
M.L.
Serial Number
D3-3386
Date
25 May '44
80
100
120
140
160
180
200
STD. C.F.M. AIR METERED @ 630 MM. Hg ABS. PRESS. & 70°F

Figure 33



ROTAMETER PRESSURE CORRECTION
630
Pressure — MM. Hg
Figure


FISCHER & PORTER COMPANY
HATBORO, PENNA
Temperature Correction Factor Curve For Motors Calibrated With Gas at 70°F
Drawn by
U.E.
Date
17 Feb '44
FIGURE 35
TEMPERATURE - DEGREES FAHRENHEIT
CORRECTION FACTOR
100
150
200
250
300
350

ORIFICE CALIBRATION $3\frac{1}{2}$' in $10\frac{1}{2}$" duct

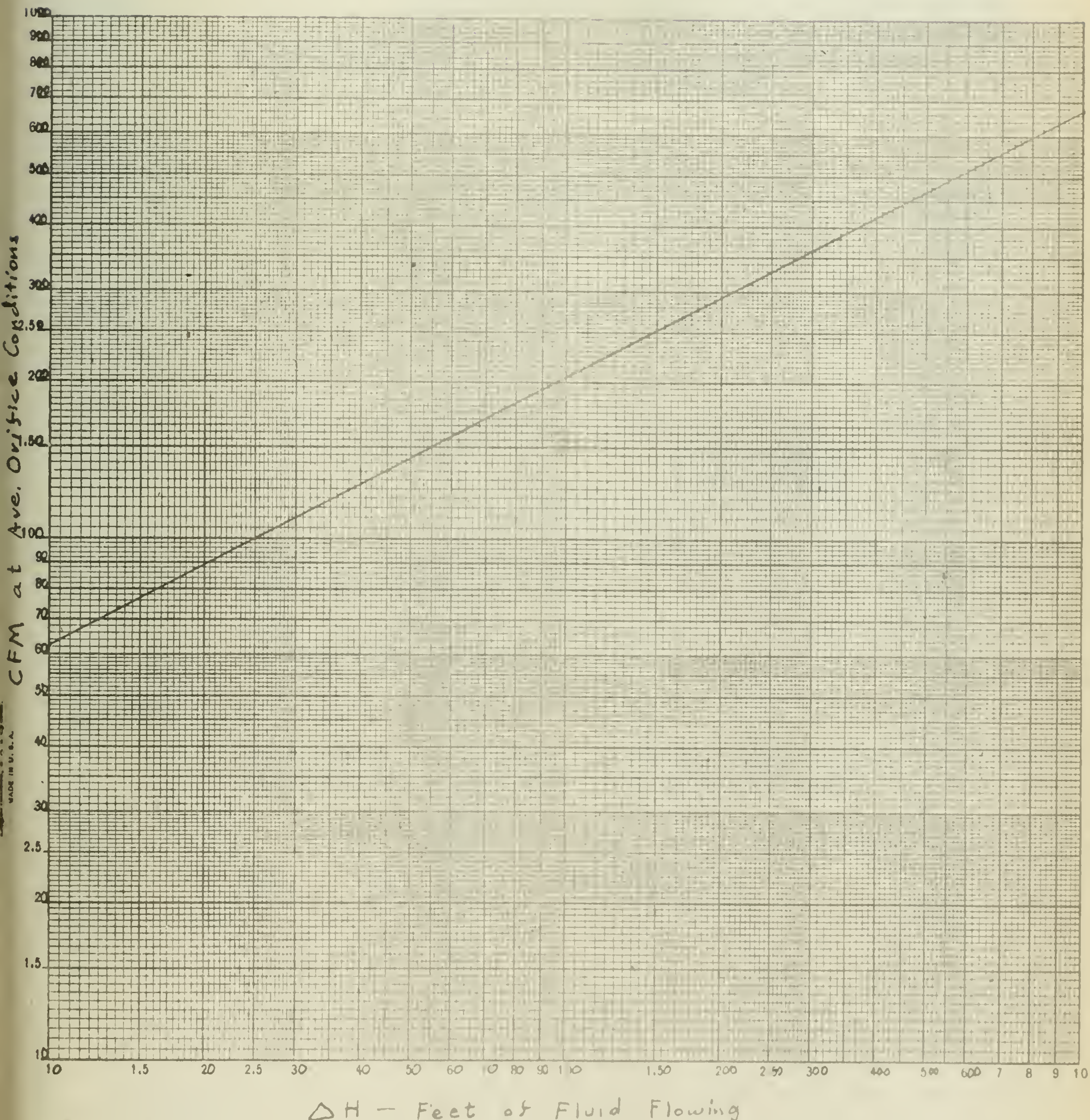


Figure 36

May 30, 1944
RWT

NOMOGRAPH FOR 3.5" AIR ORIFICE

DIRECTIONS: CONNECT BAR PRESS. B WITH STATIC PRESS. P. FROM INTERSECTION a ON REF. LINE x DRAW LINE TO ORIFICE ΔH ON M_1. FROM INTERSECTION b ON REF LINE y DRAW LINE THRU TEMP. T TO REF. LINE z. FINALLY, DRAW LINE FROM INTERSECTION c ON z, THRU ORIFICE ΔH AS NOTED NOW ON M_2. INTERSECTION ON V IS DESIRED AIR RATE AT 630 MM, 70°F

EXAMPLE: GIVEN B = 620 MM, P = 2 IN. H_2O
$M_1 = M_2 = 5$ IN. H_2O, T = 100°F
SOL'N: V = 346 C.F.M.

$$V = \frac{142.8\,M^{0.505}(B-1.07P-0.035M)^{0.495}}{T^{0.495}}$$

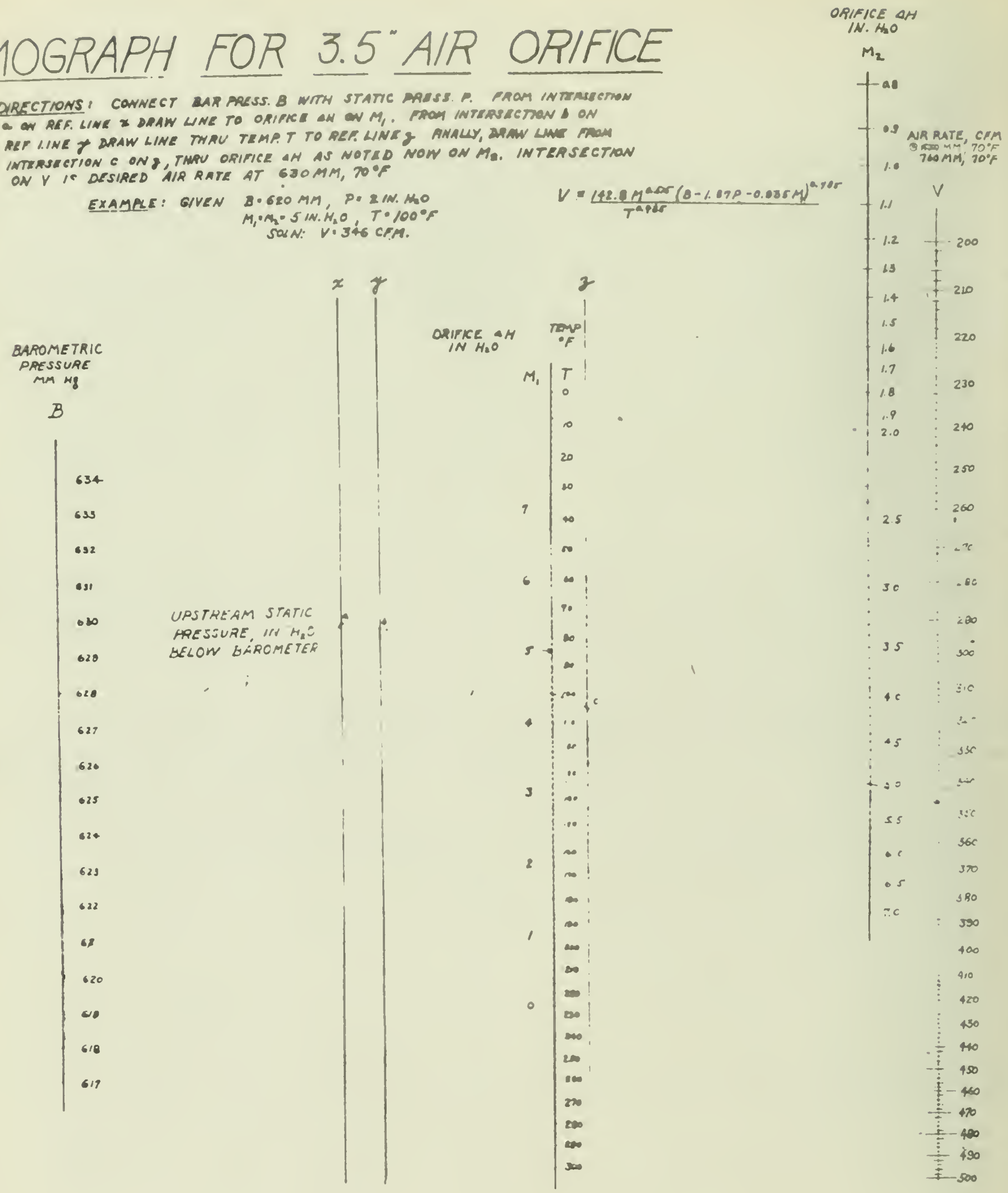


Figure 37

C. Solar Charts

1. Angle of declination versus cosine angle of incidence.

This plot was made up for the collector tilt (β) = 27° and latitude (ϕ) = 40°. Lines for 0, ±0.5------±6.5 hours from solar noon were plotted for the entire year. Calculation of one point should suffice in showing the relations used.

Nomenclature used in following sections:

θ_T = angle of incidence of direct sunlight on tilted surface

θ_Z = angle of incidence of direct sunlight on horizontal surface

δ = angle of declination of sun

ϕ = latitude = 40°

β = angle of tilt of collecter from the horizontal toward the equator = 27°

ω = hour angle, degrees (15° per hour from noon)

Angles of declination were obtained from the <u>Solar Ephemeris and Polaris Tables</u> for 1944

Angles of incidence of direct sunlight have been calculated by the following equations: (9)

$$\cos \theta_Z = \sin \phi \sin \delta + \cos \phi \cos \delta \cos \omega$$

If an artificial latitude ($\phi - \beta$) is substituted for ϕ, $\cos \theta_Z$ becomes $\cos \theta_T$ from which θ_T can be found.

$$\cos \theta_T = \sin (\phi - \beta) \sin \delta + \cos (\phi - \beta) \cos \delta \cos \omega$$

For δ = -22°, ±5.5 hrs. from solar noon

ω = 5.5 x 15 = 82.5°

$$\cos \theta_T = \sin (40-27) \sin -22 + \cos (40-27) \cos -22 \cos 82.5$$

$$= (0.225 \times -0.375) + (0.974 \times 0.927 \times 0.131)$$

$$= -0.0844 + 0.1182 = 0.034$$

$\cos \theta_T$ is plotted against δ (Figure 38, Appendix)

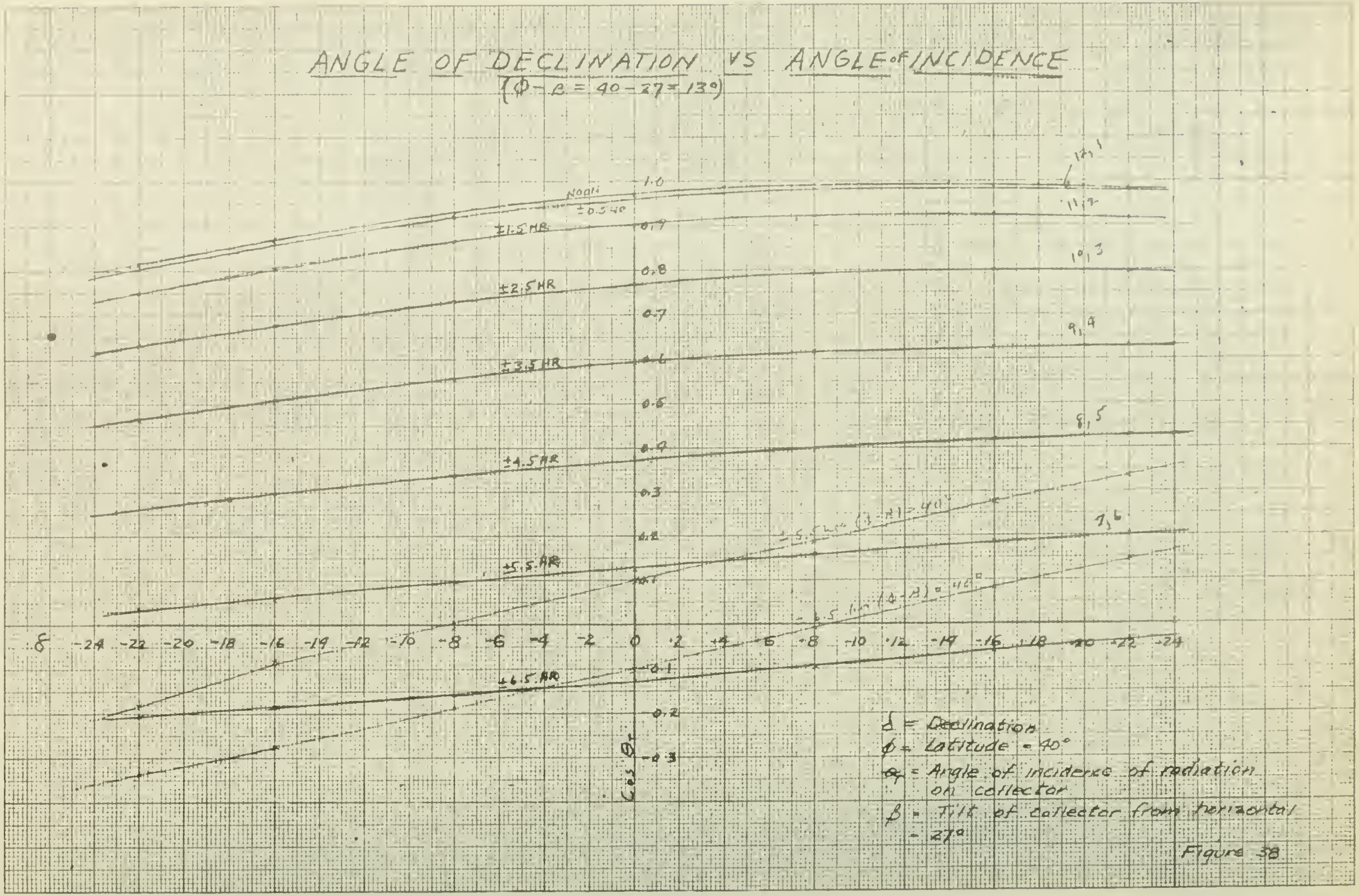

ANGLE OF DECLINATION VS ANGLE OF INCIDENCE
(ϕ − β = 40 − 27 = 13°)
NOON
±0.5 HR
±1.5 HR
±2.5 HR
±3.5 HR
±4.5 HR
±5.5 HR
±6.5 HR
12, 1
11, 2
10, 3
9, 4
8, 5
7, 6
δ
Cos θr
δ = Declination
ϕ = Latitude = 40°
θr = Angle of incidence of radiation on collector
β = Tilt of collector from horizontal = 27°

Figure 38

2. Daily interval of exposure to sunlight.

a. pyrheliometer (Figure 39, Appendix)

$\cos \theta_z = \sin\phi \quad \sin\delta \quad + \cos\phi \quad \cos\delta \quad \cos\omega$

$\theta_z = 90^\circ$ at first and last exposure

$\cos \theta_z = 0$

$\sin\phi \quad \sin\delta \quad = - \cos\phi \quad \cos\delta \quad \cos\omega$

$$\cos\omega = -\frac{\sin\phi}{\cos\phi} \tan\delta \quad ; \quad \phi = 40^\circ$$

$$= -\frac{0.643}{0.766} \tan\delta \quad = -0.84 \tan\delta$$

At $\delta = -23^\circ$

$\cos\omega = (-0.84)\ (-0.425) = 0.357$

$\omega = 69.1^\circ$

$$\pm \text{ Hrs. from noon} = \frac{69.1^\circ}{15^\circ/\text{Hr.}} = 4.61 \text{ Hrs.}$$

Time = 4:37 PM and 7:23 AM

b. collecter

$\sin(40-27) \sin\delta \quad + \cos(40-27) \cos\delta \quad \cos\omega \quad = 0$

$\cos\omega = -0.231 \tan\delta$

At $\delta = -23^\circ$

$\cos\omega = (-0.231)\ (-0.425)$

$= 0.0982$

$\omega = 84.4^\circ$

$$\pm \text{ Hrs.} = \frac{84.4^\circ}{15^\circ/\text{Hr.}} = 5.62 \text{ Hrs.} \pm \text{ noon}$$

Time = 5:37 PM and 6:23 AM.

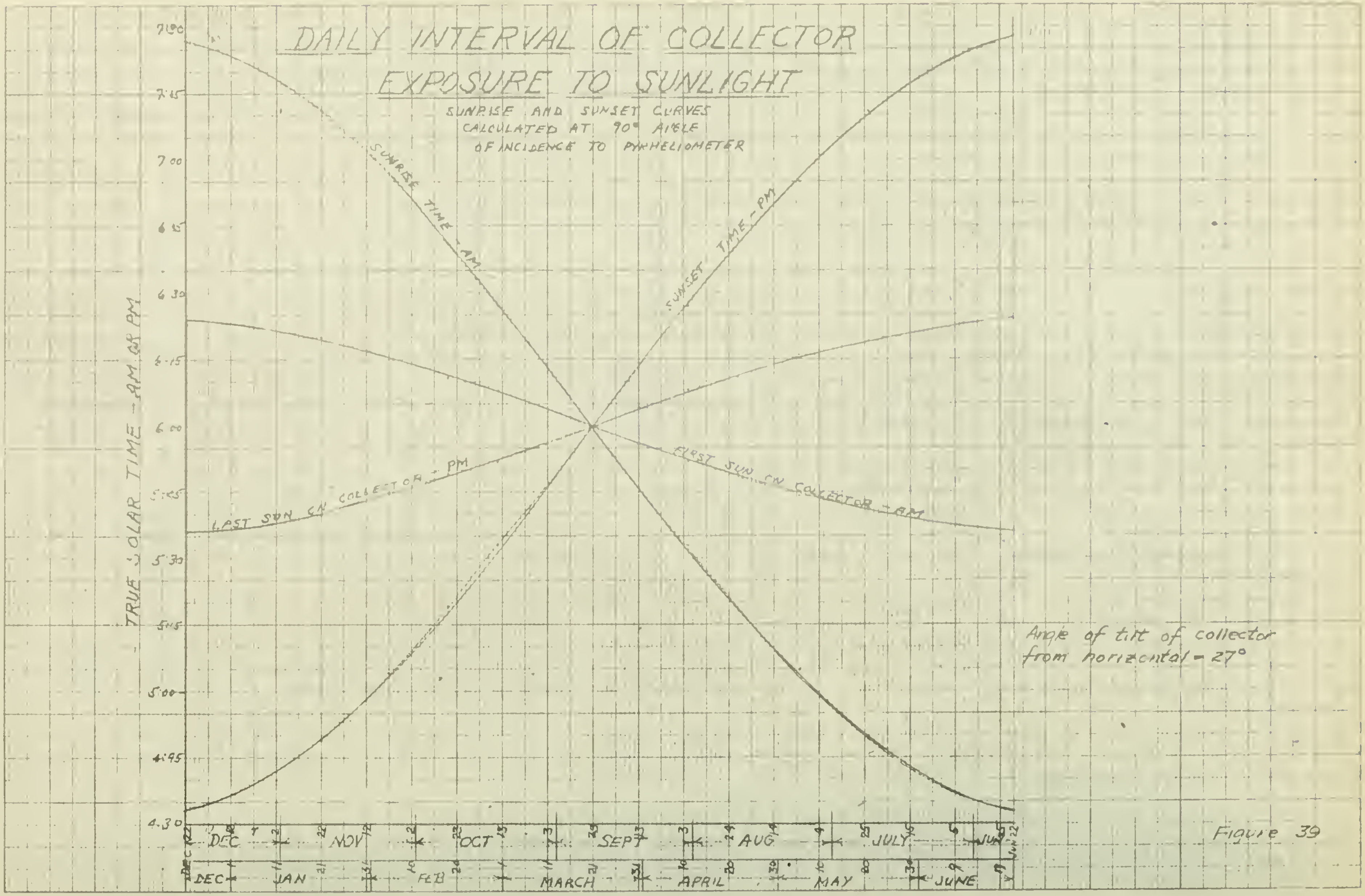

DAILY INTERVAL OF COLLECTOR
EXPOSURE TO SUNLIGHT
SUNRISE AND SUNSET CURVES
CALCULATED AT 90° ANGLE
OF INCIDENCE TO PYRHELIOMETER
TRUE SOLAR TIME - AM OR PM
7:30
7:15
7:00
6:45
6:30
6:15
6:00
5:45
5:30
5:15
5:00
4:45
4:30
SUNRISE TIME - AM
SUNSET TIME - PM
FIRST SUN ON COLLECTOR - AM
LAST SUN ON COLLECTOR - PM
DEC
NOV
OCT
SEPT
AUG
JULY
JUN
DEC
JAN
FEB
MARCH
APRIL
MAY
JUNE
Angle of tilt of collector
from horizontal = 27°

Figure 39

3. Correction factor, $R = \frac{\cos \theta_T}{\cos \theta_z}$

The correction factor must be used to convert the value for the quantity of radiation received by the pyrheliometer (horizontal surface) to that received on a tilted surface. ($\beta = 27^\circ$)

$$R = \frac{\cos \theta_T}{\cos \theta_z}$$

For $\delta = +8^\circ$; ± 5.5 hrs. from solar noon.

$\cos \theta_T = 0.155$

$\cos \theta_z = 0.19$

$$R = \frac{0.155}{0.19}$$

$$= \underline{\underline{0.816}}$$

A plot (Figure 40) of these factors was made and is shown in the appendix.

4. Reflected or transmitted radiation from 3 plates.

From Hottel's (:-) study (Figure 42) and from our plot of declination versus cosine of angle of incidence (Figure 38) the following data can be obtained:

At δ of 0°; ± 4.5 Hrs. from noon

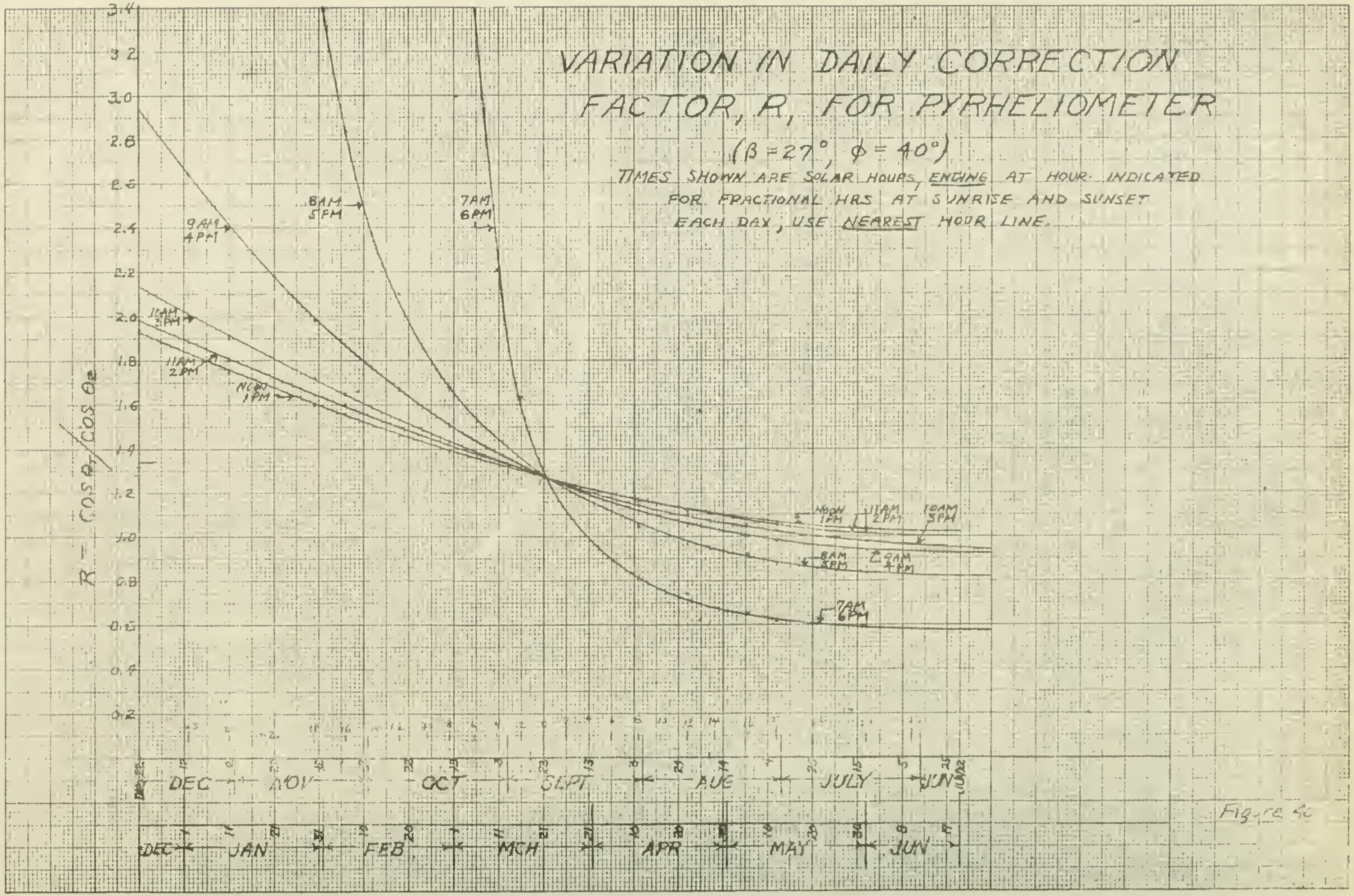
$\cos \theta_T = 0.37$

$\theta_T = 68.3^\circ$

Transmittance = 0.74 from Figure 42

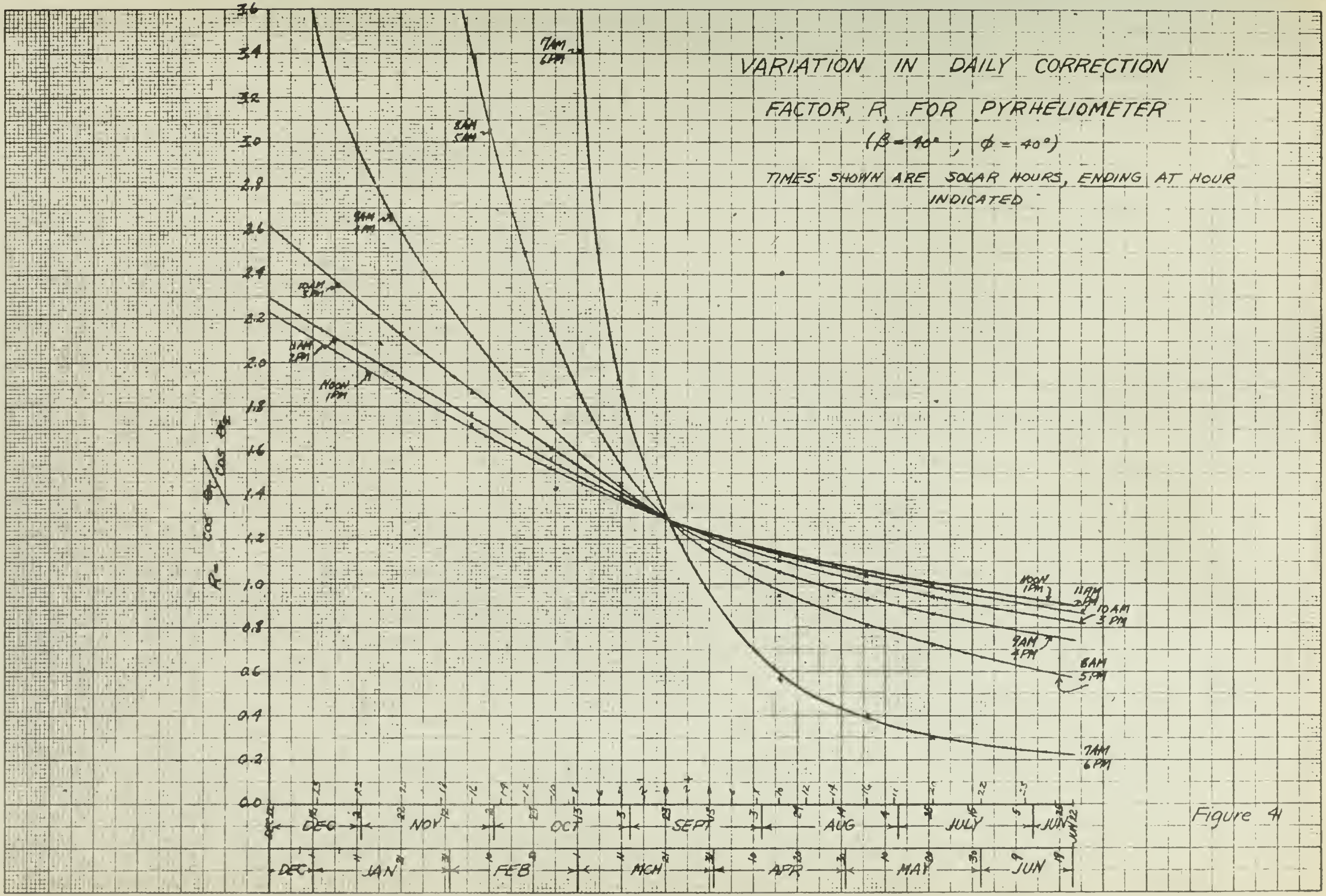
Reflectivity = 1-0.74 = 0.26

A plot (Figure 43) of the reflectivity versus time of year is shown in appendix.



VARIATION IN DAILY CORRECTION
FACTOR, R, FOR PYRHELIOMETER
(β = 27°, φ = 40°)
TIMES SHOWN ARE SOLAR HOURS, ENDING AT HOUR INDICATED
FOR FRACTIONAL HRS AT SUNRISE AND SUNSET
EACH DAY, USE NEAREST HOUR LINE.
R = Cos θ_T / Cos θ_H
3.4
3.2
3.0
2.8
2.6
2.4
2.2
2.0
1.8
1.6
1.4
1.2
1.0
0.8
0.6
0.4
0.2
7AM 6PM
8AM 5PM
9AM 4PM
10AM 3PM
11AM 2PM
NOON 1PM
DEC
NOV
OCT
SEPT
AUG
JULY
JUN
DEC
JAN
FEB
MCH
APR
MAY
JUN

Figure 5c


VARIATION IN DAILY CORRECTION
FACTOR, R, FOR PYRHELIOMETER
(β = 40°, φ = 40°)
TIMES SHOWN ARE SOLAR HOURS, ENDING AT HOUR INDICATED
7AM 6PM
8AM 5PM
9AM 4PM
10AM 3PM
11AM 2PM
NOON 1PM
3.6
3.4
3.2
3.0
2.8
2.6
2.4
2.2
2.0
1.8
1.6
1.4
1.2
1.0
0.8
0.6
0.4
0.2
0.0
DEC
NOV
OCT
SEPT
AUG
JULY
JUN
JAN
FEB
MCH
APR
MAY
Figure 4

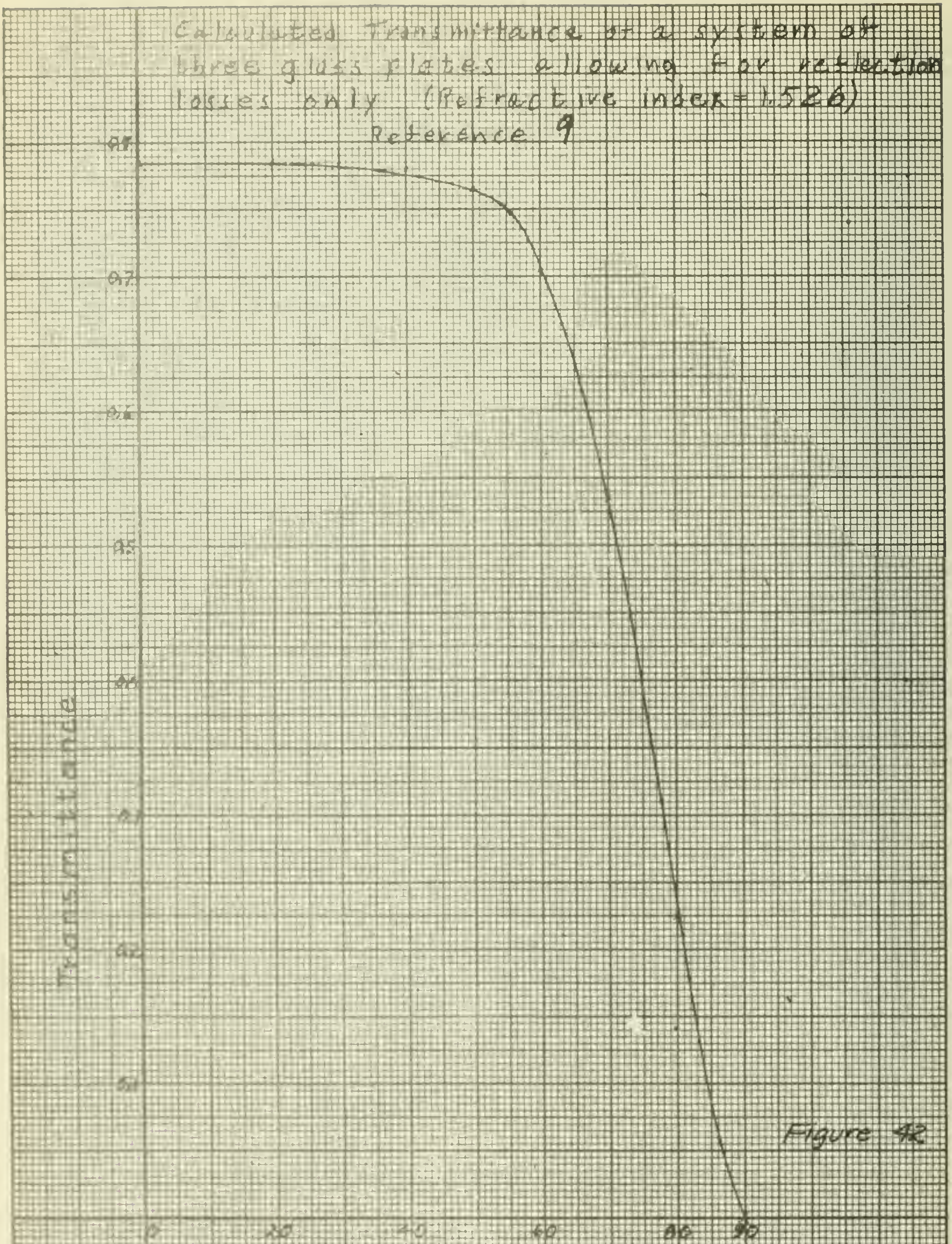

Calculated Transmittance of a system of three glass plates allowing for reflection losses only (Refractive index = 1.526)
Reference 9
0.7
Transmittance
0
20
40
60
80
90
θ, Angle of incidence, Degrees
RWH

Figure 42

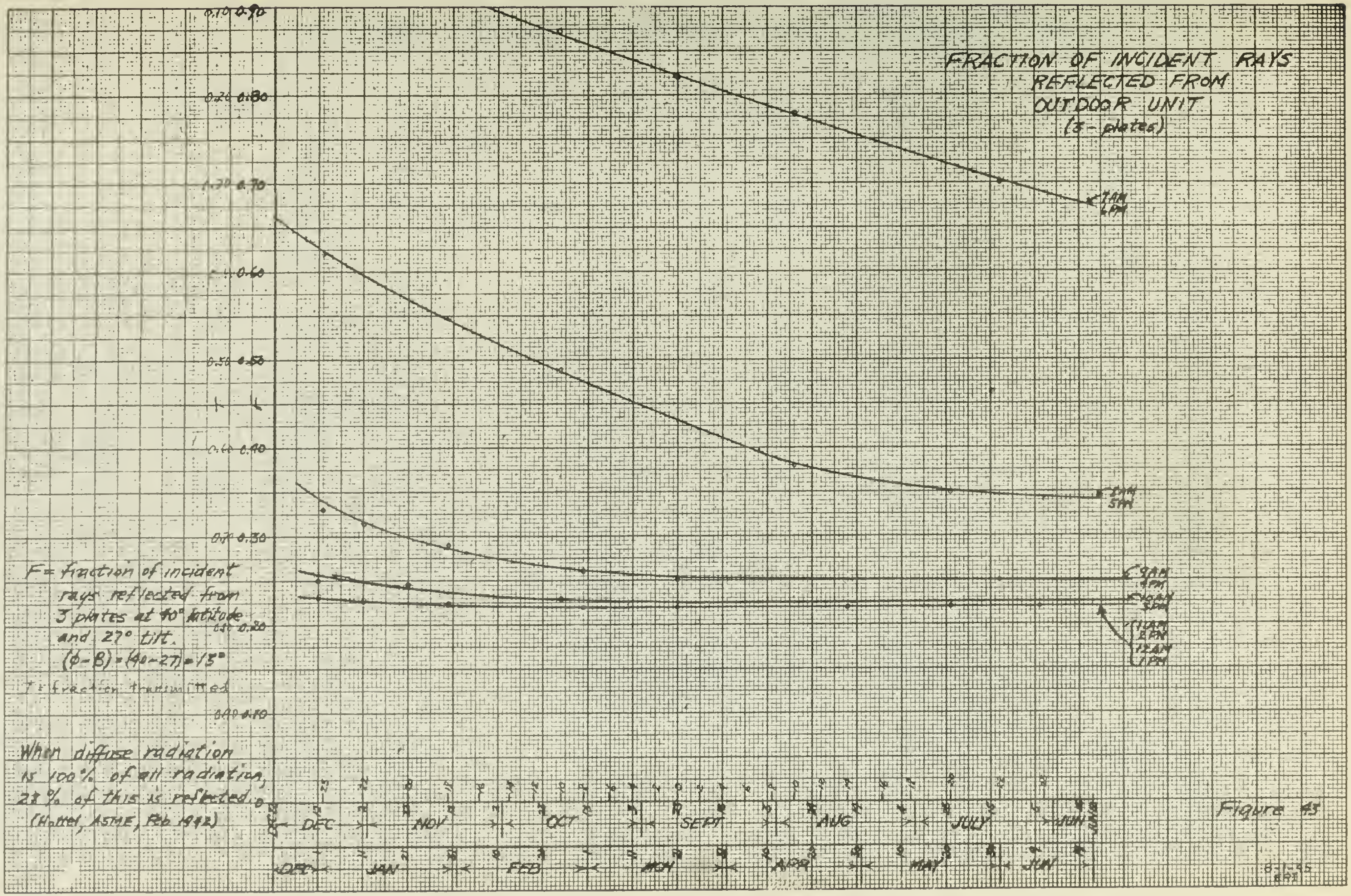
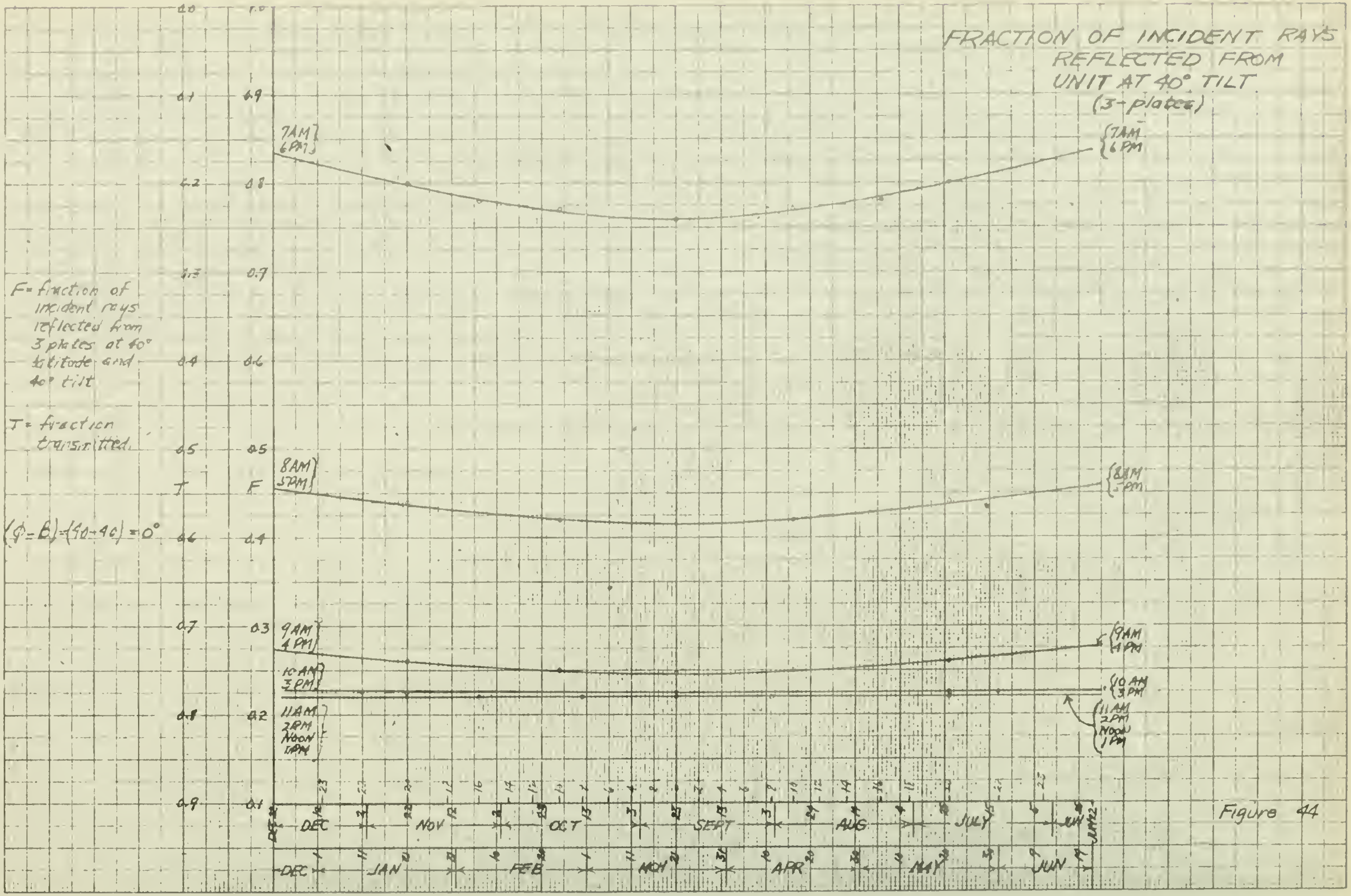

FRACTION OF INCIDENT RAYS
REFLECTED FROM
OUTDOOR UNIT
(3 - plates)
7AM
5PM
8AM
4PM
9AM
3PM
10AM
2PM
11AM
1PM
12AM
1PM
F = fraction of incident
rays reflected from
3 plates at 40° latitude
and 27° tilt.
(φ - β) = (40 - 27) = 13°
T = fraction transmitted
When diffuse radiation
is 100% of all radiation,
23% of this is reflected.
(Hottel, ASME, Feb 1942)
0.10 0.90
0.20 0.80
0.30 0.70
0.40 0.60
0.50 0.50
0.60 0.40
0.70 0.30
0.80 0.20
0.90 0.10
DEC
NOV
OCT
SEPT
AUG
JULY
JUN
JAN
FEB
MCH
APR
MAY

Figure 45



FRACTION OF INCIDENT RAYS
REFLECTED FROM
UNIT AT 40° TILT
(3-plates)
F = fraction of incident rays reflected from 3 plates at 40° latitude and 40° tilt
T = fraction transmitted
$(\phi - B) = (40 - 40) = 0°$
7AM 6PM
8AM 5PM
9AM 4PM
10AM 3PM
11AM 2PM NOON 1PM
F
T
DEC
NOV
OCT
SEPT
AUG
JULY
JUN
JAN
FEB
MCH
APR
MAY

Figure 44

II. Calculation of Results of Run

Calculations for a typical run will be shown for the hour ending at solar noon. The following data from run O-50 made on October 12, 1944, will be used:

Entrance air temperature	92°F.
Exit air temperature	184°F.
Bulk air temperature	67°F.
Coverglass temperature	123°F.
Blower temperature	109°F.
Solar Input	61.5 $\frac{\text{cal}}{\text{Cm}^2\ \text{Hr.}}$
Barometer	625 mm. Hg.
Area of heat collection, black area	209 sq. feet

Rate data (secured at 4:00 P.M.)

Rotameter static pressure	0.4" H_2O
Rotameter temperature	93°F.
Rotameter reading	107 mm.
Blower temperature	94°F.
Blower static pressure	21 mm. Hg.

The temperatures mentioned above (except the temperatures given with rate data) are mean values obtained over the period from eleven o'clock until noon. They are the arithmetic mean values for the several readings taken during the hour.

A. Air Rate-Rotameter

From the calibration curve for the rotameter, a reading of 107 mm indicates that 91.5 standard (760 mm Hg. - 70°F.) c.f.m. of air if measured at 630 mm. of Hg absolute pressure and 70°F. would be flowing. Since the air was measured at an absolute pressure of (625 mm Hg.- 0.4"H_2O and at a temperature of 93°F., pressure and temperature corrections must be applied to the 91.5 standard c.f.m. Correction factors for pressure and temperature were obtained from Figures 34 and 35 . Appendix

$$\text{Rate} = 91.5 \times 0.996 \times 0.979 = 89.2 \text{ c.f.m. (760 mm - 70°F.)}$$

$$\text{Rate at blower conditions} = 89.2 \times \frac{760}{(625-21)} \times \frac{(460 + 94)}{(460 + 70)}$$

$$= 117 \text{ c.f.m. (604 mm - 94°F.)}$$

Since the blower displaces the same volume of air constantly, the mass of air depends on the temperature and pressure at the blower. The rate calculated above is an instantaneous value based on data taken at four o'clock. An average rate for the hour is calculated using an hourly mean value of the blower temperature rather than the instantaneous temperature recorded when the rate was measured.

Actual rate at noon = $117 \times \frac{(460 + 70)}{(460 + 109)} \times \frac{(625-21)}{760}$

$= 86.7$ c.f.m. (760 mm - 70°F.)

Two other methods were used for determining air rates. A 3.5 inch air orifice was used for which a nomograph (described previously) was made. Run O-17 may be used to explain this method.

Barometer	629.9 mm Hg.
Orifice static pressure	0.35" H_2O
Pressure drop across orifice	1.73" H_2O
Orifice temperature	79°F.

Using the above data, a value of 217 c.f.m. (760 mm - 70°F.) was obtained from the nomograph. The hourly rates were obtained from the instantaneous value above by the same method described under the section on the rotameter.

The other method involved the dilution of CO_2 injected into the stream of air. Run O-1 will be used to demonstrate this method.

Barometer	24.6 inches Hg.
Static pressure, CO_2 line	26.2 inches Hg.
Pressure drop	6.4 inches Hg.
Percent CO_2	1.0%
Temperature, CO_2 line	41°F.

Total pressure = 24.6 inches Hg.
Static pressure = 26.2 " " .
CO_2 line pressure = 50.8 inches Hg = 1291 mm Hg.

Density of CO_2 $= \dfrac{44 \frac{\#}{\text{mole}}}{359 \frac{\text{ft}^3}{\text{mole}}} \times \dfrac{50.8}{29.9} \times \left(\dfrac{460 + 32}{460 + 41}\right)$

$= 0.204 \dfrac{\#}{\text{ft}^3}$

Pressure drop $= \dfrac{6.4\text{' Hg}}{12} \times \dfrac{13.6 \times 62.4 \, \#/\text{ft}^3 \text{ Hg.}}{0.204 \, \#/\text{ft}^3 \, CO_2}$

$= 2210$ ft. of CO_2

From calibration curve (20), c.f.s. of CO_2 = 0.0215

c.f.m. of air $= 0.0215 \dfrac{\text{ft}^3 CO_2}{\text{sec}} \times 60 \dfrac{\text{sec}}{\text{min}} \times \dfrac{1291}{760} \times \dfrac{530}{501} \times \dfrac{99 \text{ parts air}}{1.0 \text{ part } CO_2}$

$= 229 \dfrac{\text{ft}^3\text{air}}{\text{min}}$ (760 mm - 70°F.)

B. Cover Plate Temperature

Calculation of the cover plate temperature is as follows:

Indicated temperature = 192°F.
Instrument temperature = 86°F.
Temperature difference = 192 - 86 = 106°F.
From chart (Figure 32) actual temperature = 123°F.

C. Heat Recovered

$$\text{Heat Recovered} = \frac{\text{CFM}(760-70)}{\frac{\text{ft}^3}{359\text{mole}}} \times \frac{492}{530} \times \frac{29\frac{\#}{\text{mole}} \times 60\ \frac{\text{min}}{\text{Hr}}}{209\ \text{ft}^2} \times 0.242\frac{\text{Btu}}{\#^{\circ}\text{F}} \times \Delta t^{\circ}\text{F}.$$

$$q_R \frac{\text{Btu}}{\text{Hrft}^2} = \text{CFM}(760-70) \times \frac{492 \times 29 \times 60 \times 0.242}{530 \times 359 \times 209} \times \Delta t$$

$$q_R = \text{CFM}(760-70) \times \Delta t \times 0.00522$$

$$q_R = 86.7 \times (184-92) \times 0.00522$$

$$= 41.7\ \frac{\text{Btu}}{\text{Hrft}^2}$$

D. Heat Input

Heat Input = Solar Input x R

Where: $R = \dfrac{\cos \theta_T}{\cos \theta_Z}$

θ_T = angle of incidence of direct sunlight on a tilted surface.

θ_Z = angle of incidence of direct sunlight on a horizontal surface. (Zenith angle)

R is obtained from Figure 40 Appendix and equals 1.39 at noon on October 12.

The solar input to a horizontal surface was measured continuously by an Eppley fifty-junction pyrheliometer and recorded by a Leeds-Northrup recording potentiometer. The chart records were integrated with a polar planimeter which was calibrated to give a reading in calories per square centimeter per hour.

$$q_I = \text{Solar Input} \frac{\text{cal}}{\text{cm}^2\text{Hr}} \times \left(\frac{2.54 \times 12}{252\ \frac{\text{cal}}{\text{Btu}}}\right)^2 \frac{\text{cm}^2}{\text{ft}^2} \times R$$

$$= \text{Solar Input} \times R \times 3.69$$

$$= 61.5 \times 1.39 \times 3.69$$

$$= 315\ \frac{\text{Btu}}{\text{Hrft}^2}$$

E. Gross Efficiency (Overall for day)

$$\text{Gross Efficiency} = \frac{\text{Total Heat Recovered}}{\text{Total Gross Heat Input}} \times 100$$

$$= \frac{358}{2056} \times 100 = 17.4\%$$

F. Net Heat Input (equals gross input minus reflection loss)

$$\text{Net } q_I = \text{Gross } q_I \times t$$

Where t = transmitted radiation obtained from Figure 43 Appendix

For noon on October 12, t = 0.78

$$\text{Net } q_I = 315 \times 0.78 = 246 \ \frac{\text{Btu}}{\text{Hr.ft.}^2}$$

G. Net Efficiency (Overall for day)

$$\text{Net Efficiency} = \frac{\text{Total Heat Recovered}}{\text{Total Net Heat Input}} \times 100$$

$$= \frac{358}{1531} \times 100 = 23.3\%$$

H. Cloud Loss

$$\text{Cloud Loss} = \frac{\text{Input with clear sky - actual input}}{\text{Input with clear sky}} \times 100$$

$$= \frac{2202 - 2056}{2202} \times 100 = 6.6\%$$

The input with clear sky is the solar input obtained on a clear day a few days before or after the indicated day.

A sample data sheet of a similar run (Run 0-39) is shown on the following page.

0-39 SPACING: 1/4" OVERLAP: 2/3 PLATE LENGTH: 48" AREA: 209 sq. ft. BAR: See Table DATE: 9-17-44

No	Section	ITEM	6	7	8	9	10	11	Noon	1	2	3	4	5	6	7	8	9	10	
			Hour Interval Ending at, Solar Time																	
1	OV	Entrance Air, °F	55	56	62	70	84	91	93	95	96	93	88	83	71	70	69	67	68	77° Av.
2	OV	Exit Air, °F	53	54	68	95	122	143	160	166	162	154	136	114	87	74	70	67	68	105° Av.
3	OV	Cover Temp, °F	51	52	62	76	89	101	106	109	107	103	96	86	75	68	66	64	--	82° Ave.
4	OV	Blower Temp, °F	57	60	72	93	110	126	136	139	139	135	123	108	86	76	73	70	--	
5	OV	Bulk Air Temp, °F	55	54	54	54	60	70	76	78	79	79	78	79	76	72	68	67	--	69° Av.
6	OV	Air Rate, CFM (760-70)	395	393	384	370	359	348	343	341	341	344	350	360	374	382	384	386	386	367 AV.
7	OV	Heat Rec, BTU/HR/FT²	0	0	11.9	48.1	71.0	94.2	119.4	126.0	116.9	109.2	87.4	58.2	31.1	8.0	2.1	0	0	883.5
8	OV	Sol. Input, CAL/CM²/HR	0.5	100	27.2	44.0	57.1	66.1	70.3	68.2	64.7	53.3	37.5	19.1	0.8	0.0	0.0	0.0	0.0	
9	OV	Ht. Input, BTU/HR/FT²	1.8	39.9	121.5	200	259	302	322	312	296	242	170	85.3	3.1	0	0	0	0	2354.6
10	OV	Efficiency, %	0	0	9.4	24.0	26.4	31.2	37.2	40.3	39.5	45.2	51.4	68.1	316	--	--	--	--	37.4%
11	A	Exit Air, °F	55	56	71	98	124	144	160	164	154	140	120	105	86	76	71	68	69	Input Net 1755
12	A	Middle Plate Air, °F	--	--	73	113	164	179	192	192	182	162	138	109	--	--	--	--	--	Eff. Net 50.4%
13	A	Top Plate, °F	51	55	71	94	111	129	138	141	140	130	115	100	79	70	69	66	66	
14	A	Bottom Plate, °F	50	58	91	140	178	209	224	221	212	195	159	110	79	69	68	65	66	
15	A	Heat Rec, BTU/HR/FT²	0	0	18.0	53.8	74.7	101.5	119.5	122.3	102.8	84.0	58.3	41.3	29.2	12.0	4.0	1.9	1.9	825.2
16	B	Exit Air, °F	52	53	67	93	121	142	161	168	168	163	147	120	89	74	69	66	67	
17	B	Middle Plate Air, °F	--	--	74	107	146	175	195	201	198	184	156	119	--	--	--	--	--	
18	B	Top Plate, °F	51	55	71	90	106	127	138	141	140	132	119	100	79	70	69	66	66	
19	B	Bottom Plate, °F	51	58	89	130	167	200	219	220	214	198	164	117	81	70	68	66	67	
20	B	Heat Rec, BTU/HR/FT²	0	0	10.0	44.2	69.2	92.3	121.0	129.2	128	124.2	107.5	69.1	35.0	8.0	0	0	0	937.7
21	C	Exit Air °F	52	53	67	93	121	142	160	165	165	158	142	117	87	73	69	66	67	
22	C	Middle Plate Air, °F	54	57	72	98	124	147	161	164	161	149	130	106	79	72	70	68	--	
23	C	Top Plate, °F	50	54	74	94	112	130	140	144	149	136	124	107	78	70	68	66	66	
24	C	Bottom Plate, °F	49	53	83	128	168	203	222	224	220	201	170	124	79	68	67	63	66	129° Av.
25	C	Heat Rec, BTU/HR/FT²	0	0	10.0	44.2	69.2	92.3	119.2	124.2	122	116.2	98.5	63.7	31.2	5.9	0	0	0	896.6
26	OV	Reflection Losses	1.8	32.3	51.	50.4	58.2	66.5	70.8	68.6	65.1	54.5	42.8	35.8	2.5	0	0	0	0	600

Sunrise:	5:51 A.M.
1st Collector Sun:	5:58 A.M.
Last Collector Sun:	6:02 P.M.
Sunset:	6:08 P.M.

Barometer	24.49	24.42	24.39
Orf. read	5.3	5.1	5.2
Orf. Stat.	0.85	0.85	0.85
Orf. Temp.	121	145	133
Blower T. °F.	118	139	132
Blower CFM	484	479	481

III. Heat Balance

The heat balance was divided into the following divisions:

- input
- recovery
- total losses
    - reflection
    - convection
    - conduction
    - re-radiation
    - unaccounted

Each item was calculated for each hour of the day and the hourly values were added to give a daily value. A sample calculation for the hour ending at solar noon on October 12, 1944, (Run O - 50) follows:

A. Total losses

gross input - recovery = total losses

$$315 - 42 = 273 \ \frac{\text{Btu}}{\text{ft}^2}$$

B. Reflection losses

reflection = gross input x (1-t)

$$= 315 \times (1-.78) = 69.$$

C. Convection losses

Using the data of Jurges (17) and Schack's (18) temperature conversion factor, the heat transfer coefficient from the cover glass to the air is found by the relation

$$h_c = 0.99 + 0.21V' \left(\frac{460 + 70}{460 + t'_{air}} \right)$$

where V' is velocity of air in feet per second over the surface. Then the total heat lost per square foot is

$$\frac{q_c}{A} = h_c \, (t'_{cover} - t'_{air}) \qquad (17)$$

For $V' = 4.4$ ft/sec*
$t'_{air} = 67°F.$
$t'_{cover} = 123°F.$

$$h_c = 0.99 + 0.21 \times 4.4 \left(\frac{460+70}{460+67} \right) = 1.92 \ \frac{\text{Btu}}{\text{Hr ft}^2 °\text{F}}.$$

$$\frac{q_c}{A} = 1.92 \times (123-67) = 107 \ \frac{\text{Btu}}{\text{Hr ft}^2}$$

* Next page

*Since wind velocity data were not available on the site of the solar heating unit at the time of tho runs, an approximation was made. Wind velocity records regularly available at Valmont Steam Power Plant four miles east of the Boulder location of the unit were corrected to local wind velocities by a factor. This factor was determined from data in 1942 when data were available locally and and at Valmont. It was found that the velocity on the campus averaged approximately 0.6 that at Valmont.

D. Conduction losses

Losses by conduction were divided into three parts: (1) side losses (2) top board; (3) floor.

$$q = \frac{KA\Delta t'}{L} \quad (17)$$

K = thermal conductivity, $\frac{\text{Btu ft.}}{\text{Hr ft}^2\,^\circ\text{F.}}$

A = area of transfer, ft^2

L = thickness

$\Delta t'$ = temperature difference, $^\circ$F.

1. Side losses per square foot of collector area.

$$A = 3(15 \times 1) = 45\ ft^2$$
$$L = 1 \text{ inch (wood)}$$
$$K = 0.12$$
$$\Delta t'_1 = \frac{\text{exit + entrance}}{2} - \text{entrance air}$$
$$= \frac{\text{exit - entrance air}}{2} = \frac{184-92}{2} = 46$$
$$q_s = \frac{0.12 \times 45 \times 46}{209 \times 1/12} = 14 \frac{\text{Btu}}{\text{Hr.ft}^2}$$

2. Top losses

$$A = 15 \times 2.1 = 31.5\ ft^2$$
$$L = 4 \text{ inches (insulation)}$$
$$\quad 1 \text{ inch (wood)}$$
$$K = 0.04 \text{ (insulation)}$$
$$\quad 0.12 \text{ (wood)}$$
$$\Delta t'_2 = \text{exit air - entrance air} = 184-92 = 92$$

$$q_t = \frac{15 \times 2.1 \times 92}{209\left(\frac{4}{12\times0.04} + \frac{1}{12\times0.12}\right)} = 2 \frac{\text{Btu}}{\text{Hr.ft}^2}$$

3. Floor losses

$A = 15' \times 18' = 270 \text{ ft}^2$
$L = 8"$ (insulation)
$\quad 2"$ (wood)
$K = 0.04$ (insulation)
$\quad 0.12$ (wood)
$\Delta t_3' = (\text{bottom plate} - 70) = (274 - 70) = 204$

$$q_f = \frac{15 \times 18 \times 204}{209\left(\frac{8}{12 \times 0.04} + \frac{2}{12 \times 0.12}\right)} = 15 \frac{\text{Btu}}{\text{Hr.ft}^2}$$

$$\text{Conduction losses} = 14 + 2 + 15 = 31 \frac{\text{Btu}}{\text{Hr.ft}^2}$$

E. Re-radiation losses

$$q/A = 0.173\, \epsilon_s' \left[\epsilon_G \left(\frac{T_G}{100}\right)^4 - \alpha_G \left(\frac{T_s}{100}\right)^4\right] \qquad (17)$$

Where $\epsilon_s' = \frac{\epsilon_s + 1}{2} = \frac{0.94 + 1}{2} = 0.97$

ϵ_s = emissivity of surface

ϵ_G = emissivity of surrounding air $\simeq 1$

α_G = absorptivity of surrounding air $\simeq 1$

T_G = absolute gas temperature, °R.

T_s = absolute surface temperature, °R.

$$q/A = 0.173 \times 0.97 \left[\left(\frac{123 + 460}{100}\right)^4 - \left(\frac{67 + 460}{100}\right)^4\right]$$
$$= 64 \frac{\text{Btu}}{\text{Hr.ft}^2}$$

F. Summation - Daily totals

input - recovery = losses

$2056 - 358 = 1697 \frac{\text{Btu}}{\text{ft}^2}$

		% of total losses
reflection	525	31.0
convection	577	34.0
conduction	220	12.9
re-radiation	363	21.4
unaccounted	24	0.7

IV. Calculation of Predicted Performance of Collector for House Heating (experimental house unit)

The solar heat trap on Dr. G. O. G. Löf's house is at a tilt of 27° with the horizontal and has an area of 463 square feet. From the plot of net efficiency versus air rate obtained from studies made on the laboratory unit, (also at a tilt of 27° with the horizontal) 45% net efficiency at an air rate of 160 CFM with a 40°F. mean temperature rise were deemed optimum conditions.

A. Heat recoverable

Calculations for each hour from sunrise to sunset for every day from October, 1944, through May, 1945, were made to determine the quantity of heat recovered under the above conditions.

$$\text{Hourly Heat Recovered} = \text{Hourly Net Heat Input } \frac{\text{Btu}}{\text{Ft}^2} \times 0.45 \times 463 \text{ ft}^2$$

B. Heat necessary

1. Degree-day method

The heat necessary was calculated using the degree-day method. A temperature of 65°F. was the reference. For a day with an average temperature (arithmetic average of maximum and minimum temperature) of 42°F:

$$\text{Degree-days} = (65 - t'_{av}) \times \frac{\text{No. of Hours}}{24};\ t'_{av} < 65.$$

$$= (65-42) \times 1 = \underline{\underline{23}}$$

For calculation of heat necessary during the daylight hours alone, the same method was used with one exception. The average temperature was an arithmetic average temperature calculated from continuous chart-recorded temperatures for the daylight hours only. The above relation was used for the calculation of degree-days.

2. Heat required per degree day.

The heat required per degree-day was obtained from the gas consumption data for the Löf house.

$$\text{Heating value of gas} = 830 \frac{\text{Btu}}{\text{ft}^3} \text{ at } (630\text{mm}-70^\circ\text{F})$$

$$\text{Furnace input (19) } 90{,}000 \frac{\text{Btu}}{\text{Hr.}} = \frac{90{,}000}{830} = 108 \frac{\text{ft}^3}{\text{Hr.}}$$

$$\text{Input to pilot} = \underline{3} \frac{\text{ft}^3}{\text{Hr.}}$$

$$\text{Total input} \qquad 111 \frac{\text{ft}^3}{\text{Hr.}}$$

$$\text{Heat available to house (19) } 67{,}500 \frac{\text{Btu}}{\text{Hr.}}$$

$$\text{Equivalent heating value of gas} = \frac{67{,}500}{111} = 608 \frac{\text{Btu}}{\text{ft}^3}$$

The quantity of gas used for hot water heating was found by averaging the gas consumption during the summer months when house heating was unnecessary.

Gas used for water heating - 2430 $\frac{ft^3}{mo.}$

Period of gas consumption for this calculation only Sept. 11 - March 12

No. of degree-days - 4412° Days

Gas consumption - 123,700 ft^3

Gas consumption for water heating = 6 x 2430 = 14,600 ft

Gas consumption for house heating = 123,700 - 14,600 = 109,100 ft^3

Heat furnished to house = 109,100 ft^3 x 608$\frac{Btu}{ft^3}$ = 66,334,000 Btu.

Btu used/° Day = $\frac{66,334,000}{4412}$ = 15,035

3. Total heat necessary

Heat Necessary = Degree-days x Btu/° Day

= 23 x 15,035 = 345,000 Btu

C. Percentage of heating load carried by collector

1. No storage

The following example taken from the calculation for November will suffice with a short explanation.

	(1)	(2)	ONLY DAYLIGHT HOURS (3)	(4)	(5)	(6)
Date Nov.	°Days	Heat Necessary $x10^{-3}$	°Days	Heat Necessary Btu $x10^{-3}$	Heat Recovered Btu $x10^{-3}$	Heat Used From Sun $x10^{-3}$
24	20	301	7.9	119	125	119
25	35	527	14.6	220	166	166
26	37	557	15.0	226	265	226
27	33	497	13.8	208	197	197
28	31	467	13.3	200	197	197
29	26	391	10.8	162	146	146
30	40	602	19.6	296	101	101
Monthly Totals		10,390		4025	6700	3688

The method for calculating the values in columns 1 - 5 are shown previously. Column 6 is composed of the smaller value from either 4 or 5. The total of the column for the month are as shown.

$$\% \text{ load carried (Nov. 1944)} = \frac{(6)}{(2)} \times 100 = \frac{3688}{10{,}390} \times 100 = 35.5\%$$

$$\% \text{ load carried (Oct. 1944- Mar 1945, inc)} = \frac{23{,}770{,}000}{69{,}820{,}000} \times 100 = 34.0\%$$

2. One day storage

A similar table made up for one day storage is as follows:

Date Nov.	(1) °Day	(2) Heat Necessary Btu $\times 10^{-3}$	(3) Heat Recovered Btu $\times 10^{-3}$	(4) Heat Used Btu $\times 10^{-3}$
1	10	150	291	150
2	16	241	268	241
3	20	301	262	262
4	19	286	214	214
5	16	240	195	195
6	16	240	120	120
7	13	196	265	196
Monthly Totals		10,390	6700	6200

Column 4 is composed of the smaller values of either (2) or (3).

$$\% \text{ load carried (Nov. 1944)} = \frac{(4)}{(2)} \times 100 = \frac{6200}{10{,}390} \times 100 = 59.6\%$$

$$\% \text{ load carried (Oct. 1944- May 1945, inc.)} = \frac{46{,}570{,}000}{85{,}000{,}000} \times 100 = 54.8\%$$

3. Two and three day storage

Date	°Days	Heat Necessary Btu x10^{-3}	Heat Recovered Btu x10^{-3}	2 DAY FROM SUN Heat Used Btu x10^{-3}	2 DAY FROM SUN Heat Stored Btu x10^{-3}	3 DAY FROM SUN Heat Used Btu x10^{-3}	3 DAY FROM SUN Heat stored Btu x10^{-3}
Nov.							
1	10	150	291	150	291	150	258 291
2	16	241	268	241	268	241	291 268
3	20	301	262	301	229	301	258 262
4	19	286	214	286	157	286	234 214
5	16	240	195	240	112	240	208 195
6	16	240	120	232	0	240	163 120
7	13	196	265	196	69	196	87 265
Monthly Totals		10,393	6700	6790		7040	

Method of calculation

Example - Date - 3rd

2 day storage

Heat necessary was 301 x10^3 BTU and since there was 268 x10^3 BTU stored from the previous day, 33 x10^3 BTU were necessary to carry the total load, these were taken from the 262 x10^3 which were recovered for the day, leaving 229 x10^3 BTU in storage for use on the following day.

3 day storage

The heat in storage from two days previous to the third was 291 x10^3 BTU from the first and 268 x10^3 BTU from the second day. Therefore, the 291 x10^3 BTU were used first, leaving 10 x10^3 BTU to be taken from the 268 x10^3 BTU. After this was used there remained in storage 258 x10^3 BTU and 262 x10^3 BTU were added to this from the day's recovery.

Two day storage

% load carried (Nov.1944) = $\frac{6,790,000}{10,390,000}$ x 100 = 65.4%

% load carried (Oct.1944-May 1945, inc) = $\frac{49,470,000}{85,000,000}$ x 100 = 58.2%

Three day storage

% load carried (Nov.1944) = $\frac{7,040,000}{10,390,000}$ x 100 = 67.7%

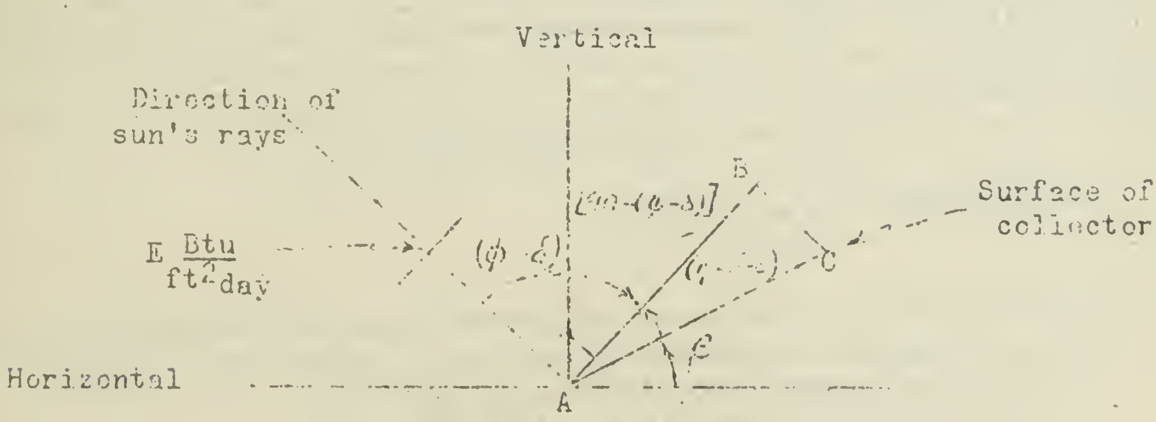
% load carried (Oct. 1944-May 1945, inc) = $\frac{51,080,000}{85,000,000}$ x 100 = 60.1%

D. Correlations for collectors at tilts other than 27°

Similar calculations as those shown for the actual collector have been made for a collector of equal area tilted 40° from the horizontal. By use of the results calculated for collectors at tilts of 27° and 40°, an attempt to approximate the radiation collected by a heat trap at any tilt has been made. The correlation shown in Figure 7 covers the solar energy obtainable over the period from October through May, the months in which collection is desired.

The method involved the derivation of an expression for the energy falling on a surface at a given tilt, β, at a latitude, ϕ=40°, and in which the change in the declination, δ, during the period is considered. Three calculations were made: (1) calculation in which the change in atmospheric transmittance with declination was neglected; (2) calculation in which the change in atmospheric transmittance with declination was included; (3) calculation in which atmospheric transmittance and relative need for the collected heat were included.

Consider in the accompanying figure that the magnitude of solar radiation is E Btu per day per square foot of surface normal to the direction of travel of the sun's ray.



latitude = ϕ = 40°
declination = δ

If atmospheric absorption is neglected, surface AB, normal to the direction of travel of sun's ray, receives E $\frac{\text{Btu}}{\text{ft}^2\text{day}}$ whereas surface AC, the collector surface, receives E cos (ϕ - δ - β) $\frac{\text{Btu}}{\text{ft}^2\text{day}}$. The total quantity of solar radiation the collector would then receive would be:

$$Q = \int_{\text{October 1}}^{\text{June 1}} E \cos(\phi - \delta - \beta)\, d\,(\text{day}).$$

If E is constant,

(1) $$Q = E \int_{\text{October 1}}^{\text{June 1}} \cos(\phi - \delta - \beta)\, d\,(\text{day})$$

Graphical integrations of the above equation for tilts, β, of 27°, 40°, 47°, 50°, 54°, and 57° were carried out and plotted as Figure 45a.

Figure 45a does not account for the variation of atmospheric transmittance with declination. Therefore, the equation was corrected to the following:

(2) $$Q = E \int_{\text{October 1}}^{\text{June 1}} \cos(\phi - \delta - \beta) \times (\text{transmittance})\, d\,(\text{day})$$

Data on the atmospheric transmittance of solar radiation as affected by declination of the sun were obtained from the report of K. W. Miller (10). Graphical integrations of the above equation for the same period and for tilts of 40°, 43°, 47°, and 50° were performed, and the results were plotted in Figure 45b.

The optimum tilt for a heat trap for maximum collection is 43°. This tilt would be perpendicular to the average direction of travel of the sun's rays. Collectors tilted so that their surfaces were parallel to the average direction of travel of the sun's rays would receive no radiation, other than a small amount of diffuse radiation. Since the curve should be symmetrical about the maximum angle, 43°, it was so drawn through the known points. The curve represents a reasonable approximation of the quantity of radiation which might be received by a collector at any tilt. For example, the radiation collected by a collector tilted 80° from the horizontal towards the equator could be approximated:

From Figure 7, $\frac{r_x}{r_{40^\circ}} = 0.73$

r_x = radiation collected by collector tilted 80° with the horizontal

r_{40° = radiation collected by collector tilted 40° with the horizontal = 64,630,000 Btu for 463 ft^2.

r_x = 0.73 x 64,630,000 Btu for 463 ft^2.
= 47,200,000 Btu for 463 ft^2.

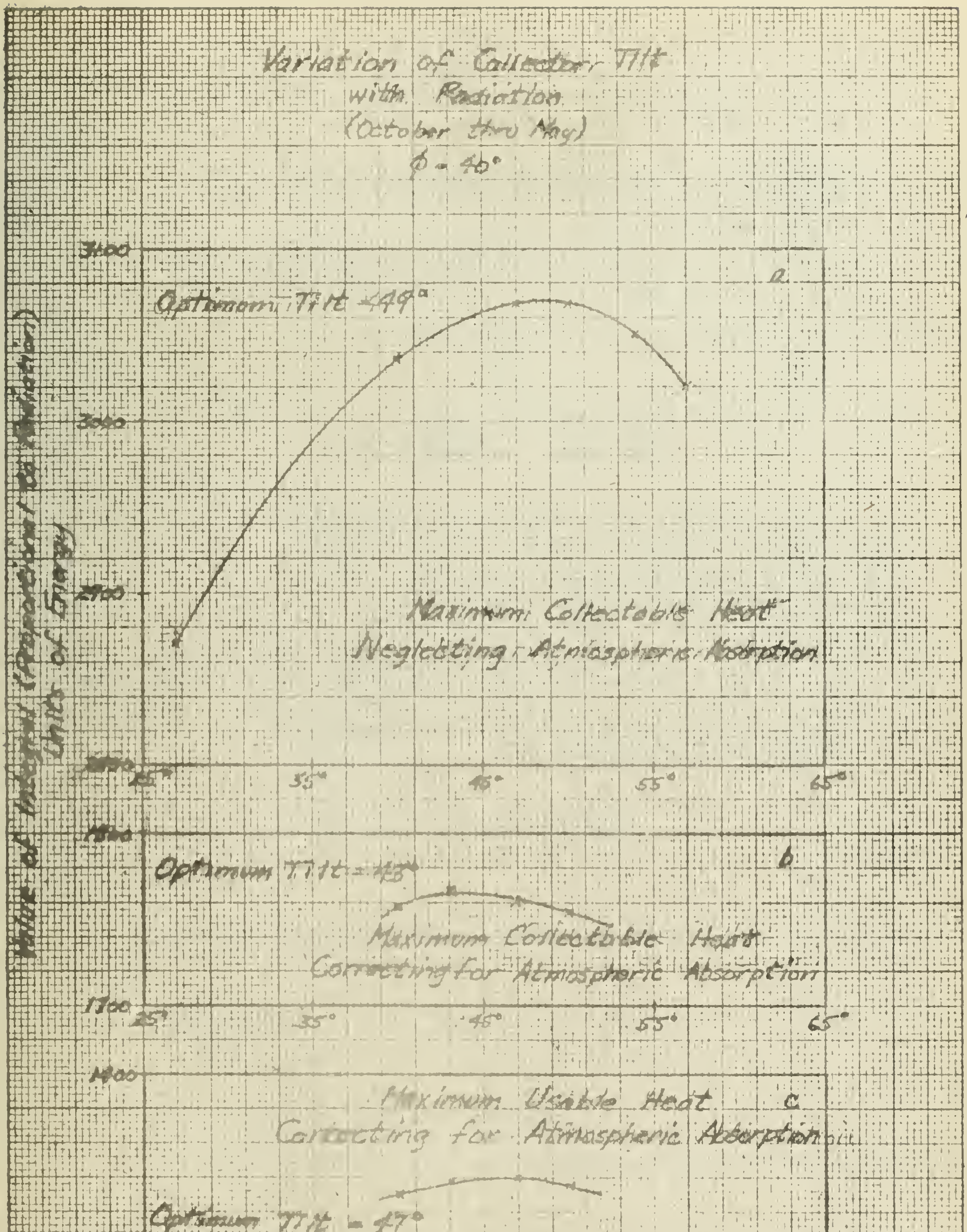

Variation of Collector Tilt
with Radiation
(October thru May)
φ = 46°
3100
a
Optimum Tilt = 49°
3000
2900
Maximum Collectable Heat
Neglecting Atmospheric Absorption
25° 35° 45° 55° 65°
1800
b
Optimum Tilt = 43°
Maximum Collectable Heat
Correcting for Atmospheric Absorption
1700
25° 35° 45° 55° 65°
1400
Maximum Usable Heat
c
Correcting for Atmospheric Absorption
Optimum Tilt = 47°
1300
25° 35° 45° 55° 65°
Angle of tilt of collector
with horizontal
Units of Energy

Figure 45

an approximation of the tilt of a heat trap for collecting the maximum heat when most desired, rather than collecting the maximum total heat for the season was obtained in the following manner. Data for the heat collected by a trap tilted at 40° and the heat used in the experimental house, if one-day storage is provided, were obtained from Table of Results II. The fraction of heat used for each month of the heating season was used as a multiplier in the graphical integration of the following equation

$$(3) \qquad Q = E \int_{\text{October 1}}^{\text{June 1}} \cos(\phi - \delta - \beta) \times \text{transmittance} \times \text{fraction of heat used } d\,(\text{day}).$$

For example, on October 15 at $\phi = 40°$ and $\beta = 40°$:

$$\delta = -8°$$

$$\text{transmittance} = 0.59$$

$$\frac{\text{heat used}}{\text{heat collected}} = \frac{4560}{9105} = 0.501$$

$Q \propto \cos(\phi - \delta - \beta) \times \text{transmittance} \times \text{fraction of heat used.}$

$Q \propto \cos(40+8-40) \times 0.59 \times 0.501$

$Q \propto \cos 8 \times 0.59 \times 0.501$

$Q \propto 0.293$ (plotted versus October 15)

Graphical integrations of this equation for tilts of 40°, 43°, 47°, and 50° were made, and the results were plotted in Figure 45c. A tilt of 47° is seen to be the optimum for collecting heat usable during this season, if the monthly fractions of collected energy actually used are the same as in the experimental house.

V. Predicted Size of Heat Storage Unit

A. One day storage

From the previous calculation (Sec. IV C 2), it was observed that an average of the high quantities of heat used from the sun over a one day period was 300,000 Btu. The maximum quantity of heat used from the sun over a one day period was 400,000 Btu. The size of storage bins large enough to store these quantities could be estimated as follows:

1. assume a value of 0.2 $\frac{\text{Btu}}{\#^{o}\text{F}}$ for the specific heat of the storage material.
2. assume an available temperature rise of (200-70) = 130°F.

Storage of 300,000 Btu

$$\frac{300{,}000\ \text{Btu}}{0.2\frac{\text{Btu}}{\#^{o}\text{F}} \times 130^{o}\text{F}} = 11{,}500 \text{ pounds of material}$$

% heating load carried = 53.51%

Storage of 400,000 Btu

$$\frac{400{,}000\ \text{Btu}}{0.2\frac{\text{Btu}}{\#^{o}\text{F}} \times 130^{o}\text{F}} = 15{,}400 \text{ pounds of material}$$

% heating load carried = 54.78%

B. Two day storage

Storage of 500,000 Btu, average high quantity of heat stored over a period of two days.

$$\frac{500{,}00\ \text{Btu}}{0.2\frac{\text{Btu}}{\#^{o}\text{F}} \times 130^{o}\text{F}} = 19{,}200 \text{ pounds of material}$$

% heating load carried = 56.52%

Storage of 740,000 Btu, the maximum quantity of heat stored over a two-day period.

$$\frac{740{,}00\ \text{Btu}}{0.2\frac{\text{Btu}}{\#^{o}\text{F}} \times 130^{o}\text{F}} = 28{,}500 \text{ pounds of material}$$

% heating load carried = 58.19%

C. Three day storage

Storage of 800,000 Btu, an average high quantity of heat stored over a period of three days.

$$\frac{800,000\ \text{Btu}}{0.2\frac{\text{Btu}}{\#\,^\circ\text{F}} \times 130^\circ\text{F}} = 30,800 \text{ pounds of material}$$

Storage of 970,000 Btu, the maximum quantity of heat stored over a three-day period.

$$\frac{970,000\ \text{Btu}}{0.2\frac{\text{Btu}}{\#\,^\circ\text{F}} \times 130^\circ\text{F}} = 37,300 \text{ pounds of material}$$

D. General Comments

From the above calculations, it is seen that two sizes of storage units have been estimated. Survey of the day-by-day collected heat that could be stored shows that storage of the maximum quantity of heat is unwarranted because the overall percentage heating load carried would not be increased sufficiently. Therefore, it was proposed to use a storage unit capable of storing an average high value.

VI. Calculation of Theoretical Performance of Unit

A. Heat balance (based on data of run O-42)

Incident radiation = 329 $\frac{Btu}{Hr ft^2}$

Radiation reflected from top surface = 0.04 x 329 = 13.2 $\frac{Btu}{Hr ft^2}$

Radiation transmitted to plate = 315.8 $\frac{Btu}{Hr ft^2}$

Radiation absorbed = 315.8 x 0.03 + $\left(\frac{47-13.2}{0.96} + \text{abs}\right)$ x 0.03= 10.6 $\frac{Btu}{Hr ft^2}$

t_2 plates (Miller) = 0.798 for a=0.03, r=0.08, t=0.89 (1 plate)

Radiation absorbed on black surface= 0.798 x 329 = 262 $\frac{Btu}{Hr ft^2}$

a_2 plates = 0.059 (10)

Radiation absorbed (total) in glass = 0.059 x 329 = 19.4 $\frac{Btu}{Hr ft^2}$

Radiation absorbed in 2nd plate = 19.4 - 10.6 = 8.8 $\frac{Btu}{Hr ft^2}$

Heat into air stream = 329 - losses

r_2 plates= 0.143 (Miller) for a = 0.03, r = 0.08, t = 0.89 (1 plate)

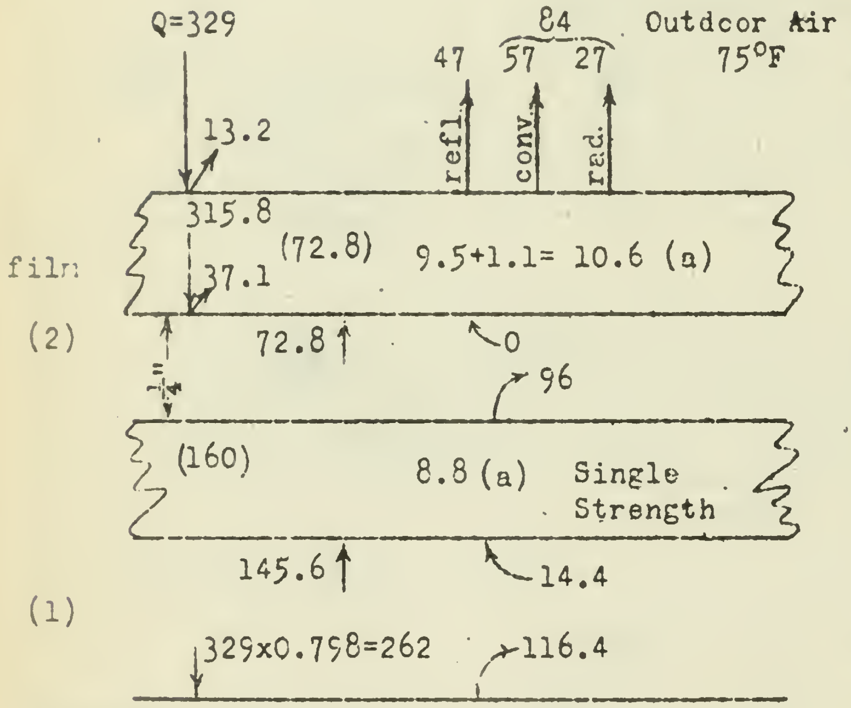
Radiation reflected = 0.143 x 329 = 47 $\frac{Btu}{Hr ft^2}$

Radiation convected from top surface = 57 $\frac{Btu}{Hr ft^2}$ (from previous calculation)

Radiation reradiated from top surface= 27 $\frac{Btu}{Hr ft^2}$ (from previous calculation)

Heat into air stream = 329 - 47 - 57 - 27 = 198 $\frac{Btu}{Hr ft^2}$

MILLER ANALYSIS - RUN 0-42
(5th calculation)



Assumed q	Assumed S	T, °F	Calculated $\bar{T}$, °F	Calculated q	Calculated S
		100			
		101.3			
96	1.0	158	115.8	93	1.1
		160.7			
102	0.78		186.3	105	0.76
		243			

B. Calculations based on Miller's analysis

1. Assume Q_n & s_n $\quad s_2 = 1.0 \quad Q_2 = 96$ Btu/Hrft^2

Cover temp = 100°F $\quad s_1 = 0.78 \quad Q_1 = \underline{102}$ Btu/Hrft^2

$\qquad\qquad\qquad\qquad\qquad\qquad\qquad\qquad 198$ Btu/Hrft^2

Film 2: $-q_1 = \frac{s-1}{2s} Q_2 = \frac{1-1}{2} 96 = 0$

$$q_2 = \frac{s+1}{2s} Q_2 = \frac{2}{2} \times 96 = 96 \frac{Btu}{Hrft^2}$$

Film 1: $-q_1 = \frac{s-1}{2s} Q_1 = \frac{0.78-1}{2 \times 0.78} \times 102 = 14.4 \frac{Btu}{Hrft^2}$ (from air to plate)

$$q_2 = \frac{s+1}{2s} Q_1 = \frac{0.78+1}{2 \times 0.78} \times 102 = 116.4 \frac{Btu}{Hrft^2}$$ (from plate to air)

2. Run heat balance around film for each film. (See diagram)

3. Calculate temperatures of plates.

Cover temperature (exp.) = 100°F

Cover temperature (bottom) $T_0 = 100 + \frac{Q_c + A/2}{K}$; Q_c = heat conducted through plate, $\frac{Btu}{Hrft^2}$

$$T_0 = 100 + \frac{72.8+5.3}{60}$$

$$= 100 + 1.3$$

$$= 101.3°F$$

A = heat absorbed in plate, $\frac{Btu}{Hrft^2}$

K = Conductance of glass = $60 \frac{Btu}{Hr\,°F\,ft^2}$

Middle plate (top) $T_{00} = 100\left[\left(\frac{T_0+460}{100}\right)^4 + \frac{Qr}{0.172\frac{e}{2-e}}\right]^{1/4} - 460$;

$$= 100\left[\left(\frac{101.3+460}{100}\right)^4 + \frac{72.8}{0.1555}\right]^{1/4} - 460$$

$$= 158°F$$

Q_r = heat radiated between plates, $\frac{Btu}{Hrft^2}$

Middle plate (bottom) $T_0 = 158 + \frac{160+4.4}{60}$

$$= 160.7\ °F$$

e = emmissivity of glass surface = 0.95

T_0 = temperature of bottom of upper plate, °F.

Black plate (top) $T_{00} = 100\left[\left(\frac{160.7+460}{100}\right)^4 + \frac{145.6}{0.1555}\right]^{\frac{1}{4}} - 460$

$$= 243\ °F$$

62088

4. Calculate average air temperatures, $\bar{T}$.

Film 2: $\bar{T}_2 = -\frac{17}{70}\ s\ (T_{oo}-T_o) + \frac{T_{oo}+T_o}{2}$

$$= -\frac{17}{70} \times 1 \times (158 - 101.3) + \frac{158 + 101.3}{2}$$

$$= 115.8^oF$$

Film 1: $\bar{T}_1 = -\frac{17}{70} \times 0.78\ (243-160.7) + \frac{243+160.7}{2}$

$$= 186.3^oF$$

5. Calculate k of air at $\bar{T}$;

Film (2) $k = \frac{0.00273}{\bar{T} + 684}\left(\frac{\bar{T}+460}{492}\right)^{3/2} \times 3600$

$$= \frac{0.00273}{115.8+684}\left(\frac{115.8+460}{492}\right)^{3/2} \times 3600$$

$$= 1.55 \times 10^{-2}\ \frac{Btu\ ft}{Hr\ ft^2\ ^oF}$$

Film (1) $k = \frac{0.00273}{186.3 + 684}\left(\frac{186.3 + 460}{492}\right)^{3/2} \times 3600$

$$= 1.7 \times 10^{-2}\ \frac{Btu\ ft}{Hr\,ft^2\ ^oF}$$

6. Calculate critical spacing, 2ho and s.

Film (2) $2h_o = \frac{2k\ (T_{oo}-T_o) \times 12}{Q_n}$ $\qquad s = \frac{2h}{2h_o} = \frac{0.25}{2h_o}$

$$= \frac{2 \times 1.55 \times 10^{-2} \times 56.7 \times 12}{96} \qquad = \frac{0.25}{0.23}$$

$$= 0.23 \text{ inches} \qquad = 1.1$$

Film (1) $2h_o = \frac{2 \times 1.7 \times 10^{-2} \times 82.3 \times 12}{102}$ $\qquad S = \frac{0.25}{0.33}$

$$= 0.33 \text{ inches} \qquad = 0.76$$

7. Calculate Q_1 and Q_2. Plot $\bar{T}$ at centers of sections vs position and extrapolate

$$Q_n = \frac{\Delta\bar{T}_n}{\Sigma\Delta\bar{T}} \times Q_T$$

Where $\Sigma\Delta T$ is total drop from the hotter end of plate to the colder end, ^oF.
$\Delta\bar{T}_n$ is drop from the hotter end of section to the colder end, ^oF.
Q_T is total heat picked up in air stream; $\frac{Btu}{Hr\,ft^2}$

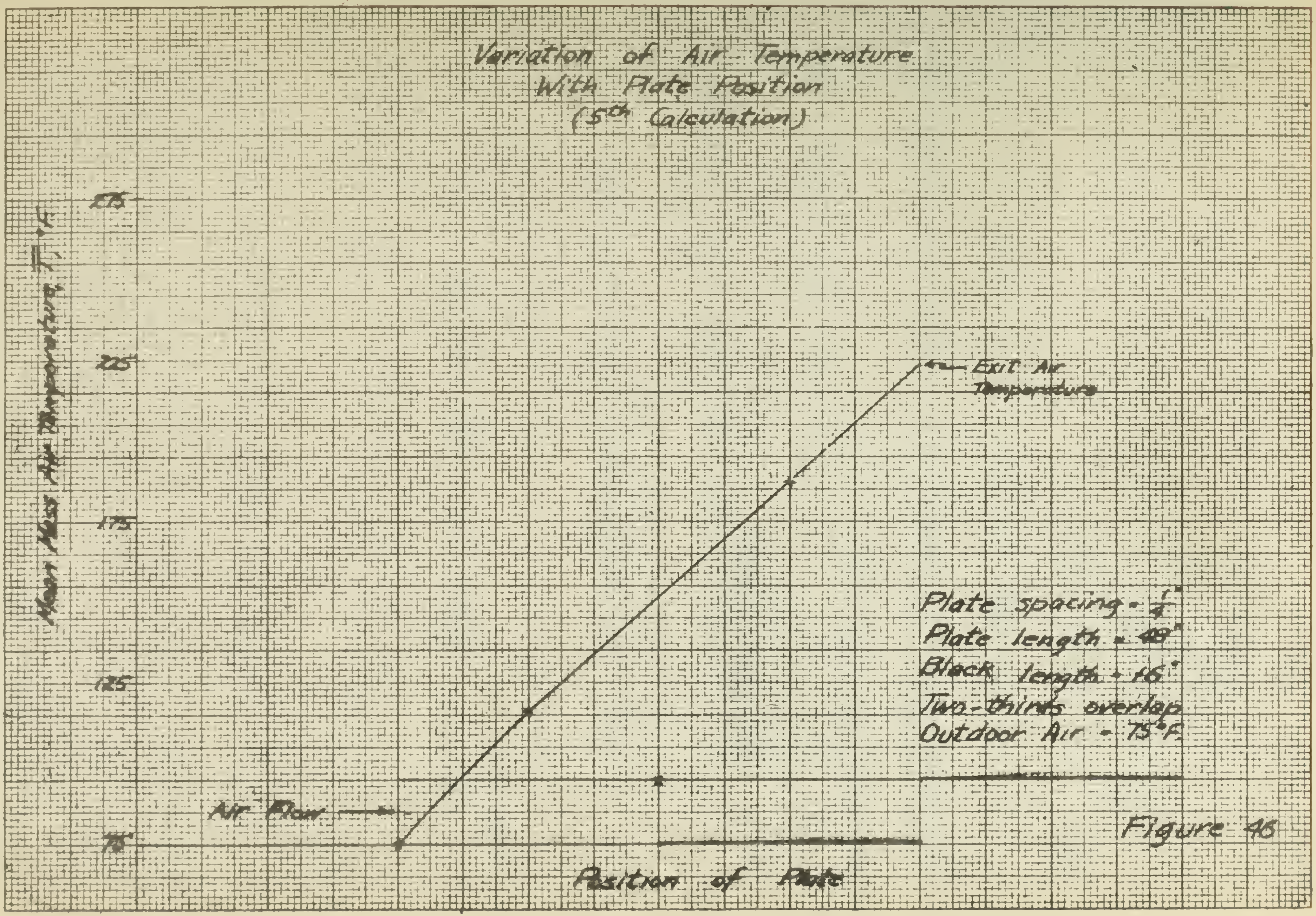

Variation of Air Temperature
With Plate Position
(5th Calculation)
Mean Mass Air Temperature T, °F
275
225
175
125
75
Exit Air Temperature
Plate spacing = 1/4"
Plate length = 48"
Black length = 16"
Two-thirds overlap
Outdoor Air = 75°F
Air Flow
Position of Plate

Figure 46

Film 2 $\quad Q_2 = \dfrac{\overline{\Delta T_2}}{\Sigma \Delta T} Q_T$

$$= \frac{224-154}{224-75} \times 198$$

$$= \frac{70}{149} \times 198$$

$$= 93 \frac{\text{Btu}}{\text{Hr ft}^2}$$

Film 1 $\quad Q_1 = \dfrac{154-75}{224-75} \times 198$

$$= 105 \frac{\text{Btu}}{\text{Hr ft}^2}$$

8. Recalculate steps 1 - 7 using new Q and s until comparable with the values assumed in step 1.

9. Calculate rate of air flow

$$\text{CFM} = \frac{Q_T \times W \times L}{0.238 \Sigma \Delta \overline{T} \rho}$$

; W= heated length of plate = 32"

L= Width of plate = 36"

ρ= Density of air = 0.075 $\frac{\#}{\text{ft}^3}$ at 70°F - 760 mm Hq.

$$= \frac{198 \times 32/12 \times 36/12}{60 \times 0.238 \times 149 \times 0.075}$$

$= 9.95$ c.f.m./(section)(two films)

$= \frac{9.95}{2}$ c.f.m./(section)(film) x 5 sections x 9 films

= 224 c.f.m.

10. Gross efficiency

$$\text{Efficiency} = \frac{\text{Output}}{\text{Input}} \times 100 = \frac{198}{329} \times 100 = 60\%$$

VII. Nomonclature

a Fraction of incident radiation absorbed by glass.
B Barcmetric pressure, mm. Hg.
c Heat capacity, Btu./#°F.
g Acceleration due to gravity, 32.17 ft./sec.^{2}
h One-half the distance from one plate to the other, units of length.
h_c Heat transfer coefficient for convection, Btu./(hr)(sq ft)(°F).
h_o One-half the critical distance from one plate to the other, units of length.
i Incident light or radiation.
k Thermal conductivity, (Btu)(ft)/(hr)(sq ft)(°F).
M Pressure drop across orifice, inches H_2O.
n Subscript referring to the number of plates.
P Pressure, #/ft^2 or mm. Hg.
P' Upstream static pressure on orifice, inches H_2O.
q_c Heat lost by convection, Btu/(hr)(ft^2 of collector area).
q_I Heat input, Btu/(hr)(ft^2 of collector area).
q_f Heat lost through floor of collector, Btu/(hr)(ft^2 of collector area).
q_R Heat recovered, Btu/(hr)(ft^2 of collector area).
q_s Heat lost through sides of collector, Btu/(hr)(ft^2 of collector area).
q_t Heat lost through top of collector, Btu/(hr)(ft^2 of collector area).
R Correction factor for solar radiation falling on a horizontal surface to that falling on a tilted surface equal to cosine θ_T/cosine θ_Z.
r Fraction of incident radiation reflected from glass.
s Ratio, $2h/2h_o$
t Fraction of incident radiation transmitted through glass.
t' Temperature, °F.
$\underline{T}$ Temperature of gas at a point in the film, °F.
T Mean mass temperature of gas, °F.
T_G Absolute gas temperature, °R.
T_o Upper glass plate temperature, °F.
T_{oo} Lower glass plate temperature, °F.
T_S Absolute surface temperature, °R.
V Volume of air, c.f.m. (760mm.-70°F.).
V' Velocity of air, ft./sec.
w Length of section taken, units of length.
x Distance from center of section taken, units of length.
y Distance from center of film to any point, units of length.

Greek

α_G Absorptivity of surrounding air, dimensionless.
β Angle of tilt of collector from the horizontal toward the equator degrees.
δ Angle of declination of sun, degrees.
ΔT Temperature rise of glass from one end to the other of section taken, °F.
ϵ Emissivity, dimensionless.
ϵ_G Emissivity of surrounding air, dimensionless.
ϵ_S Emissivity of surface, dimensionless.
μ Viscosity of gas, #/(ft)(sec).
ϕ Latitude, degrees.
θ_T Angle of incidence of direct sunlight on a tilted surface, degrees.
θ_Z Angle of incidence of direct sunlight on a horizontal surface, degrees.
ω Hour angle, degrees (15° per hour from noon).

VIII. Definitions

Cloud loss - $\frac{\text{input with clear sky - actual input}}{\text{input with clear sky}}$ x 100, percent.

Degree-day - a unit, based upon temperature difference and time, used in specifying the nominal heating load in winter. For any one day there exists as many degree-days as there are degrees Fahrenheit difference in temperature between the mean temperature for the day and 65°F.

Gross efficiency - (heat recovered)(100)/(gross heat input), percent.

Gross heat input - solar input x R, Btu./(hr)(ft^2).

Net efficiency - (heat recovered)(100)/(net heat input), percent.

Net heat input - gross heat input x t, Btu./(hr)(ft^2).

XI APPENDIX

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